THE APPLICATION OF COMPUTER MODELLING TO THE DESIGN OPTIMISATION OF A MICROEXCAVATOR DIGGING ARM

by

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CERTIFICATION OF RESEARCH

This is to certify that, except where specific reference is made, the work described in this dissertation is the result of the investigation of the candidate.

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W. Evans.

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Director of Studies

DECLARATION

This is to certify that neither this dissertation, nor any part of it, has been presented, or is being concurrently submitted, in candidature for any degree at any other University.

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NOTATION

Symbol	Definition	Units
A	Area	m^2
A_{XX}	Area of cross-section at location x	m^2
angtooth	Angle of applied bucket tooth force to bucket tooth	degrees
angboom	Angle between upper and lower boom	degrees
angres	Resultant angle of measured bucket tooth force to the horizontal	degrees
F	Force	N
F_K	Force at joint K	N
F_{KX}	Horizontal component of F_K for global coordinates	N
F_{KY}	Vertical component of F_K for global coordinates	N
F_{KH}	Horizontal component of F_K for local coordinates	N
F_{KV}	Vertical component of F_K for local coordinates	N
Favge	Average bucket tooth force for a range of geometric configurations	N
Fmax	Maximum bucket tooth force for a range of geometric configurations	N
Fres	Resultant measured bucket tooth force	N

Symbol	Definition	Units
f_{RAM1}	Frequency of operation of service line relief valve 1 for a range of geometric configurations	
F_{tmax}	Maximum bucket tooth force for a single geometric configuration	N
I_{XX}	Second moment of area at location X	m^4
l	length	m
M_X	Bending moment at location X	Nm
P	Pressure	N/m^2
T_x	Torsional moment about x axis	Nm
S.F.	Safety factor based on yield stress	
x	Distance of point of interest from vertical datum in local coordinates	m
X_K	Distance of point of interest K from vertical datum in global coordinates	m
y	Distance of point of interest from horizontal datum in local coordinates	m
Y_K	Distance of point of interest K from horizontal datum in global coordinates	m
σ	Shear Stress	N/m^2
σ_d	Direct Stress	N/m^2
σ_b	Bending Stress	N/m^2
σ_c	Combined Stress	N/m^2
σ_y	Material Yield Stress	N/m^2

SUMMARY

An assessment of the parameters which must be considered for the design of microexcavator digging arm and hydraulic system has been undertaken.

Fundamental mechanics and structural theories have been used to develop a structural model of the Powerfab 360WT microexcavator digging arm under static or Quasi-Static loading conditions.

Practical strain gauge tests were carried out to assess the accuracy of the computer model with satisfactory results.

A design optimisation method has been developed using a structural model integrated with a hydraulic system model. A complete computer aided design software package has been written utilising these models, enabling the design engineer to analyse the effects of design changes to the digging arm structure or hydraulic system to optimise structural integrity and performance.

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CHAPTER ONE

1 INTRODUCTION

Powerfab is a traditional engineering company designing and manufacturing microexcavators[1] originally conceived by the Company's chief executive, David John, in 1979. Microexcavators are small scale mechanical excavators capable of exerting digging forces of up to 26kn and have grown rapidly in popularity in recent years by mechanising previously expensive manual labour operations (Meadows[1]). One of their major features is their ability to access confined areas where larger machines find it impossible to work.

In the Fiercely competitive construction equipment market, Powerfab have continued to grow steadily but realised that their market lead was threatened by larger companies. The introduction of new technology was seen to provide a firmer base for future development allowing them to maintain their role as a market leader.

1.1 Initial Collaborative Work

During the autumn of 1983 Powerfab were experiencing design problems with their Mk1 Powerfab 125 Microexcavator and Dr. Evans, a Stress Analyst in the Department of Mechanical and Production Engineering, The Polytechnic of Wales was contacted to give technical advice. As a result of discussions a consultancy project was undertaken by the department. The project involved the experimental field testing of the Mk1 Powerfab 125 Microexcavator in order to obtain stress values on the structure as a number of pre-selected positions (Plate 1.1). The results were used by the Company to highlight possible zones of fracture. Design modifications were accordingly made thus giving the Company more confidence in the structural integrity of the finished product (Evans[2]).

The successful outcome of the consultancy project resulted in the formation of the first Teaching Company Programme between the Department of Mechanical and Production Engineering and Powerfab Limited, in November 1984. Experimental testing is a time consuming

PLATE 1.1

Experimental Field Testing of the Powerfab 125 Microexcavator Mk.1

and costly business and so it was decided to apply a commercial software package to the structural analysis of the new Mk2 Powerfab 125 Microexcavator. Computer modelling methods provide fast results at a fraction of the time and costs of experimental methods. The commercial software package "SAGS" or the Department's VAX 11/780 mainframe computer, readily available in the Department, was used to model the digging arm structure. A number of experimental tests were carried out to determine the accuracy of the computer model. Only one major component of the digging arm was modelled and the results showed large discrepancies, O'Brien[3]. These discrepancies in results were due to inaccurate modelling of the structure using the software package. The re-modelling of the entire digging arm structure using the package was carried out by an undergraduate engineering student. The results again showed large discrepancies between the computer predicted and experimental results, Stephens[4].

1.2 An Assessment of the Computer Modelling Package 'SAGS'

The commercial software package 'SAGS' by SDRC has a number of limitations for this particular application. The method employed by the software package is based on a stiffness matrix method (SDRC[5]). Each member of the structural model must have known stiffness properties, this presents a problem when trying to model the hydraulic cylinders [since their stiffness properties change depending on the cylinder rod extension, expansion of the hydraulic pipes and the hydraulic cylinder]. The rams therefore cannot accurately be modelled and so applied forces must be introduced along the line of the hydraulic cylinder reacting at each of the hinge pins, so three applied forces are required using this technique representing each of the hydraulic cylinders.

The package required the user to determine the X-Y-Z co-ordinates in space of all of the major joints defining the structural model, these are then set up in a data table for input to the model. A new set of co-ordinates is required for each new geometric configuration, thus the process of examining forces and stresses for a number of positions is time consuming and occupies a large amount of memory.

The presentation of output data is confusing and constant references to the program notation must be made. The results are presented as a table of numbers with no graphical output.

The package itself cannot easily be modified to perform additional calculations, if required, such as determining the point of maximum stress over the entire structure.

The use of a large mainframe computer package is not commercially viable for the company concerned. The co-operation of the department would be necessary for computer time and a great deal of familiarity with the package required before any results could be obtained - the package is not 'user friendly'.

1.3 Microcomputer Based Software Package

A survey of available microcomputer based software packages was made, similar limitations were identified with these packages. The microcomputer solution is low cost, well within the budget of the company, can be programmed in the language BBC BASIC and is also simple to operate.

It was decided to develop an in-house customised computer software package to carry out the structural design analysis and design optimisation of a microexcavator.

1.4 Previous Published Material

There is no published material specifically related to the theoretical structural analysis of a mechanical excavator. The general approach in the past has been the experimental testing of and full scale excavators using strain gauge application (Jacques[6]) or stress coating techniques (Tyrer[7]). The manufacturing of scale models can prove to be just as costly as producing a full scale prototype. Development using such models is consequently long and expensive. The stress coating method allows areas of suspected stress concentration to be identified but some prior knowledge of the stress distribution is necessary.

It is recognised that manufacturers in the market for larger excavators have developed their own or commissioned specialised computer packages for structural analysis. Company confidentiality prevents these packages however from being available for use by the smaller manufacturer and no published material is available.

CHAPTER TWO

2 DESIGN CONSIDERATIONS

This chapter describes in detail the various considerations that are made at the design stage of a microexcavator. The principles involved in the design of a microexcavator are similar to those required for the design of much larger excavator capable of digging forces of greater than 26KN.

2.1 General Layout of a Microexcavator

Microexcavators are generally small scale versions of the larger mechanical excavator. The major difference is their accessibility to confined areas when in the folded up position. The digging arm assembly consists of three major parts - the Bucket, Dipper and Boom assembly (Figure 2.1). Each component can be manoeuvred using one or more of the hydraulic cylinders. In a typical digging operation each cylinder is operated independently or in combination with the other hydraulic cylinders in order to achieve the required digging path and digging forces. The digging arm itself is connected to the upper platform of the microexcavator. The Powerfab 360WT microexcavator has a full 360° slew capability hence the upper platform is connected to the lower platform via a slew ring arrangement as shown in (Figure 2.2.) (Powerfab[8]).

2.2 Hydraulic System

The hydraulic power unit consists of a 10 HP air cooled petrol engine driving a hydraulic pump and integral cooling fan (Figure 2.3) the pump draws the oil supply from the tank and feeds the control valve block (Powerfab[8]). The control valve block consists of a bank of 5 double acting spool valves, one control valve for each of the hydraulic cylinders (known as service line). The hydraulic cylinders are double acting cylinders (Trade and Technical Press[9]) with differing full bore and annular areas. Each service line contains a 'Poppet' type relief valve to limit the hydraulic pressure in each line. This ensures that the applied load or induced load does not exceed a pre-determined limit thus preventing damage to the structure. A main line relief valve is also included in the circuit

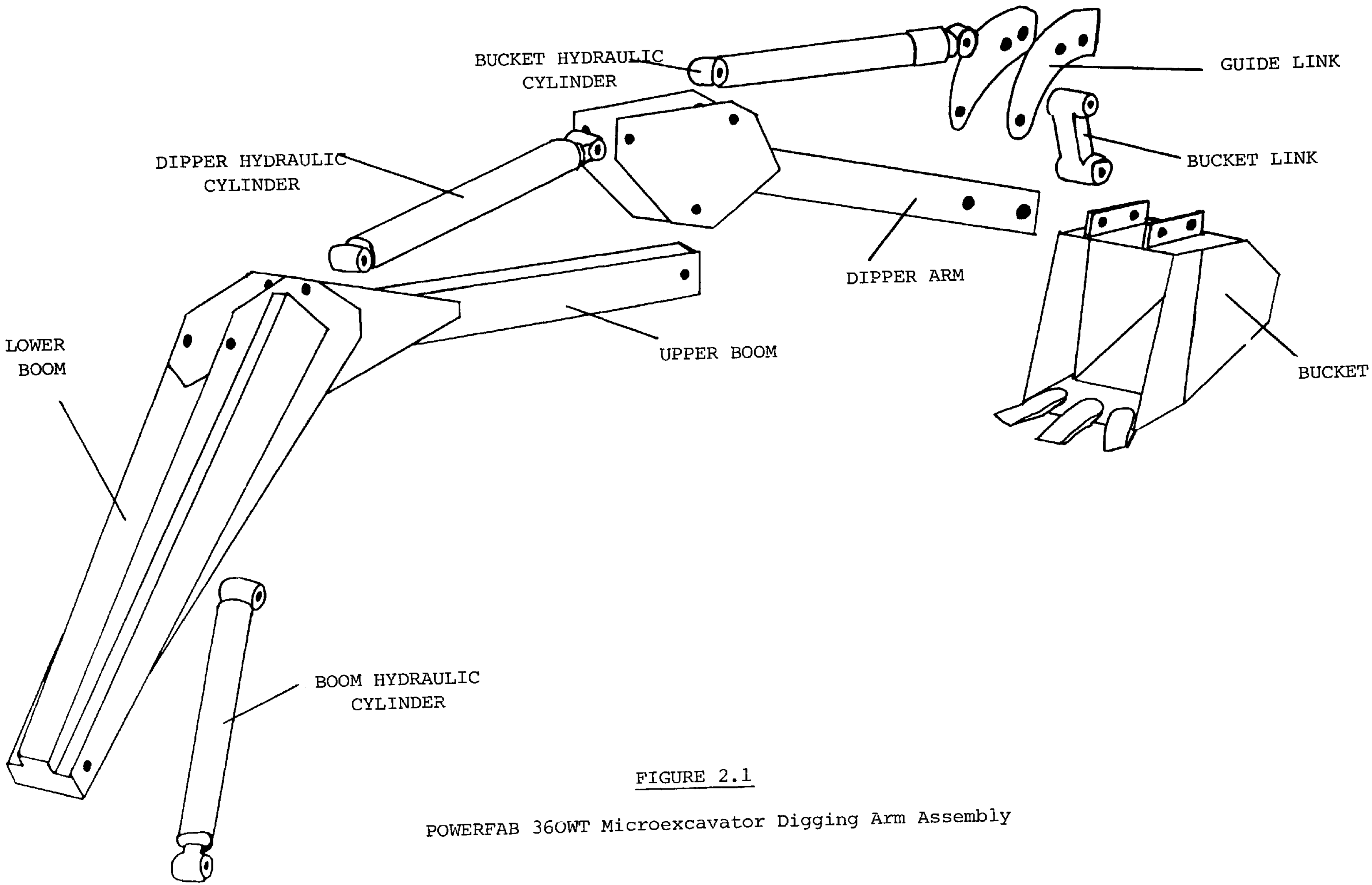


FIGURE 2.1

POWERFAB 360WT Microexcavator Digging Arm Assembly

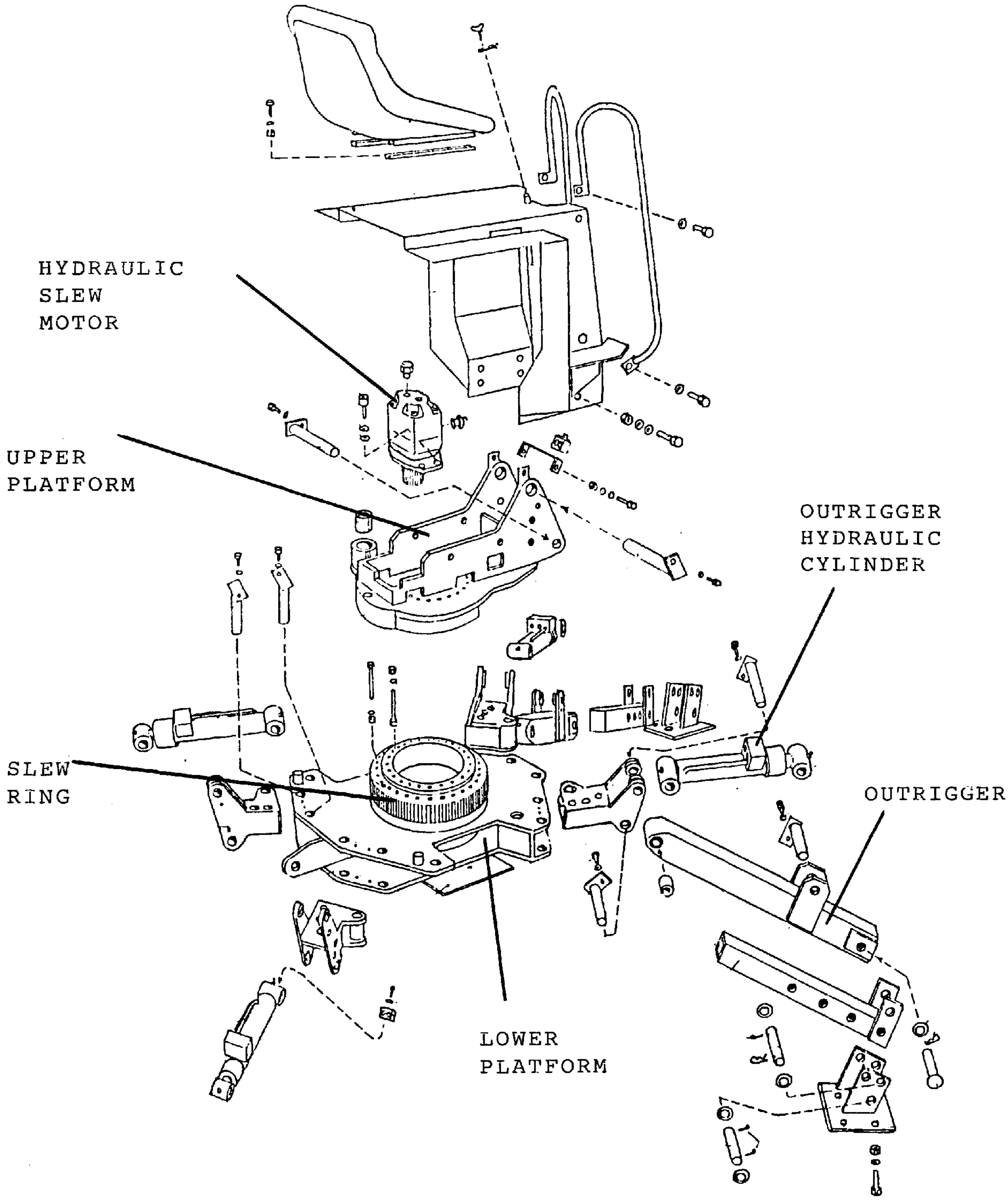


FIGURE 2.2

Exploded View of the POWERFAB 360WT Microexcavator Platform Assembly (POWERFAB[8])

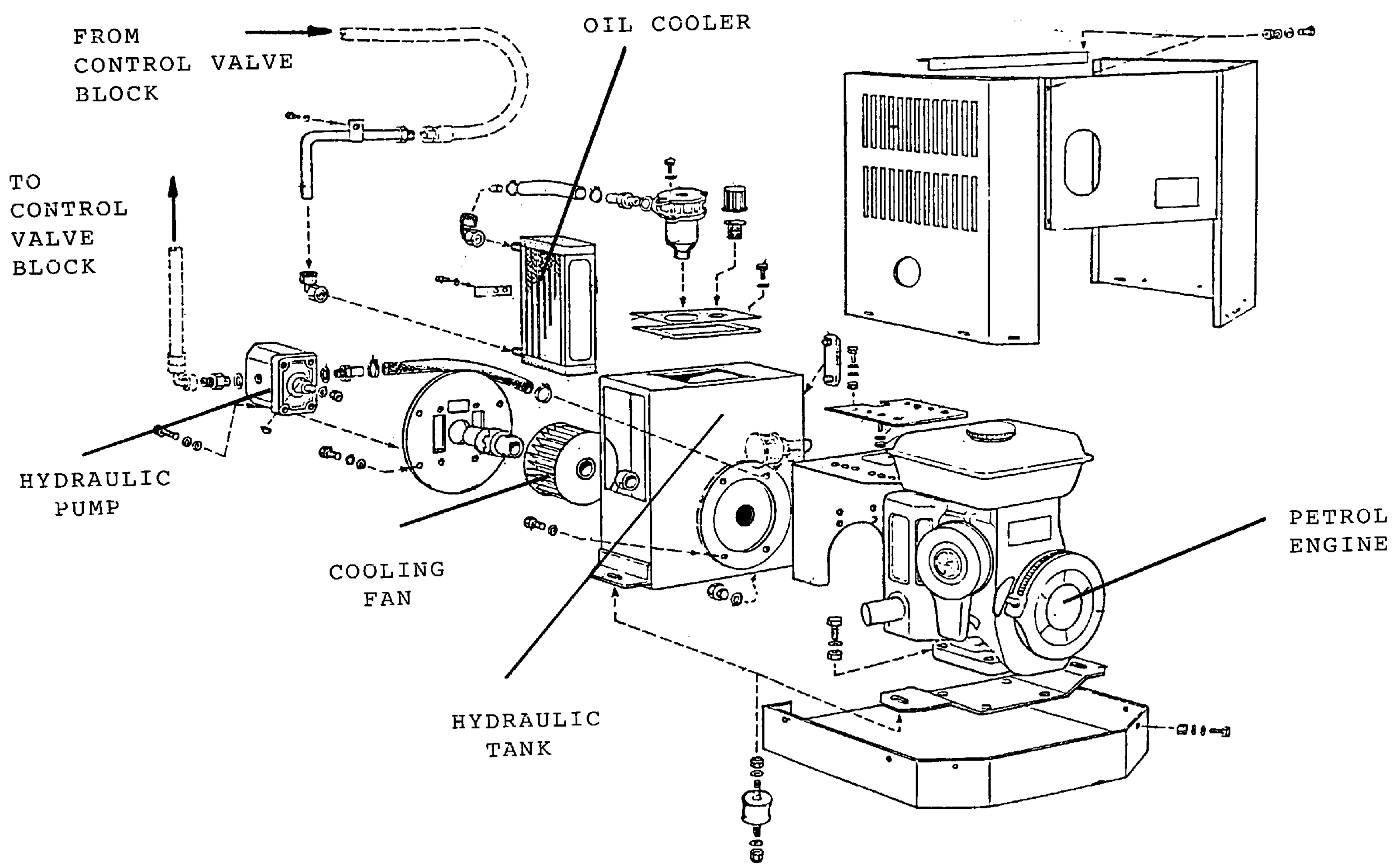


FIGURE 2.3

Exploded View of the POWERFAB 360WT Microexcavator Hydraulic PowerUnit (POWERFAB[8]) to ensure that the supply pressures to the control valve block and hence each individual service do not exceed a set pressure limit. The complete 'CETOP' diagram for the hydraulic circuit is shown in Figure 2.4 (this uses standard symbols to represent the hydraulic circuit, Trade & Technical Press [9]). Large forces can be induced when the boom and dipper are allowed to move under the forces of gravity. To limit the dynamic forces that result flow restrictors are placed in the appropriate service lines.

2.3 Digging and Lifting Stability

Under normal working conditions the microexcavator may be used for both digging and lifting operations. Before commencing either operation a stable working platform is essential, this requirement is of paramount importance when digging or lifting operations occur on unlevel ground. The Powerfab 360 Microexcavator has four hydraulic operated 'feet' known as outriggers to assist in these operations. These are lowered into the ground and the machine raised off its towing wheels to provide a stable platform (Figure 2.5), (Powerfab[10]). Some microexcavators operate in conditions where the lower platform is fixed to the 'ground'. While digging or lifting the microexcavator may become statically or dynamically unstable. The effects of dynamic stability are minimised by careful design of the hydraulic system to ensure that hydraulic cylinder operating speeds do not endanger the operator and the effects of static stability are discussed in the following sections.

2.3.1 Static Stability While Lifting

When the operator attempts to lift a heavy load (assuming that sufficient hydraulic cylinder forces are available), the rear outriggers may be lifted off the ground when the digging arm is extended (Figure 2.6). As the digging arm is brought closer into the machine the lifting capacity increases greatly and the limiting stability condition acts as a 'safety' limit to the operator, who immediately controls the digging arm into a more stable condition. The achievable lifting load is a function of the position of the digging arm in space and the available hydraulic cylinder forces.

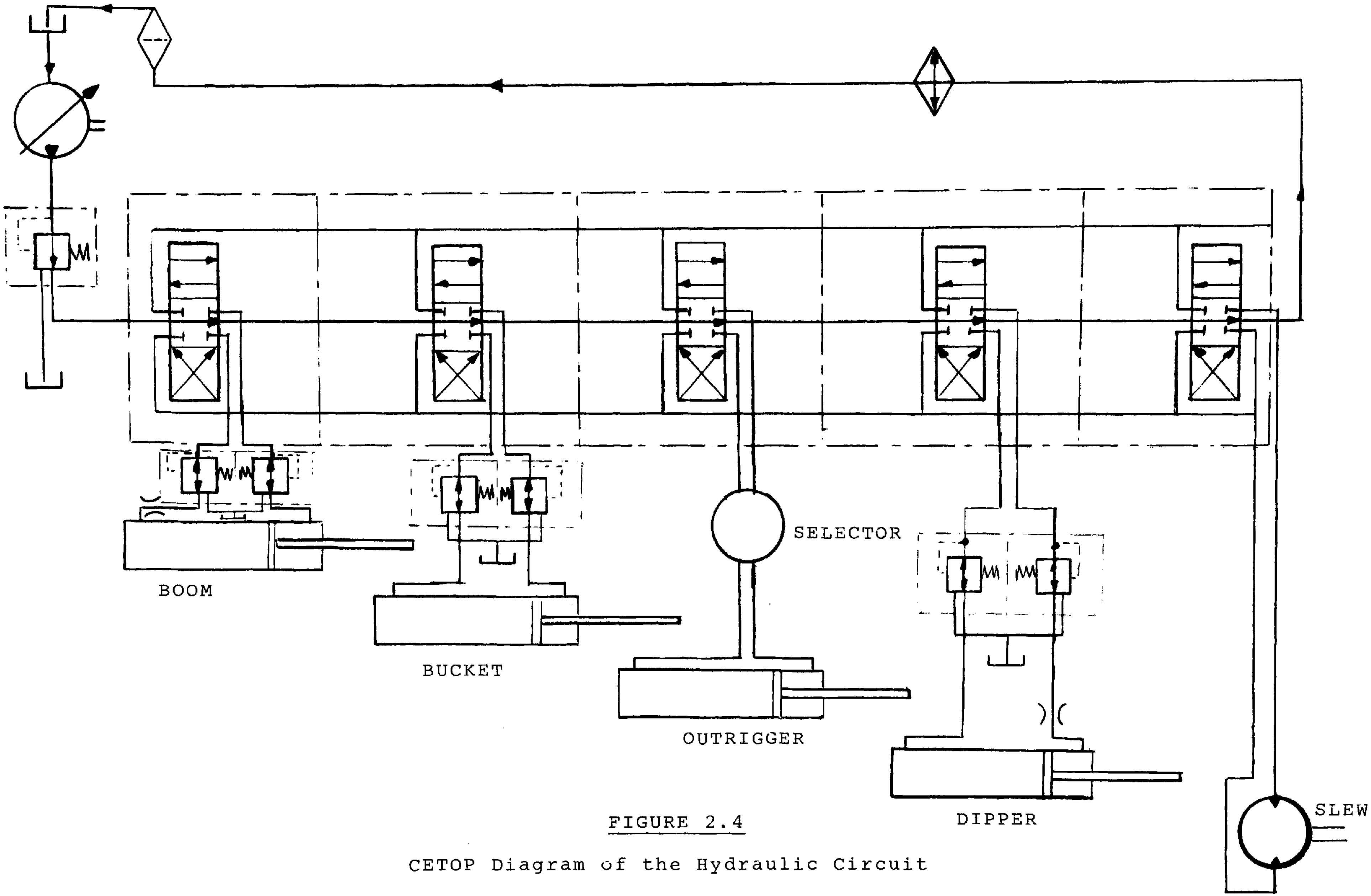


FIGURE 2.4

CETOP Diagram of the Hydraulic Circuit

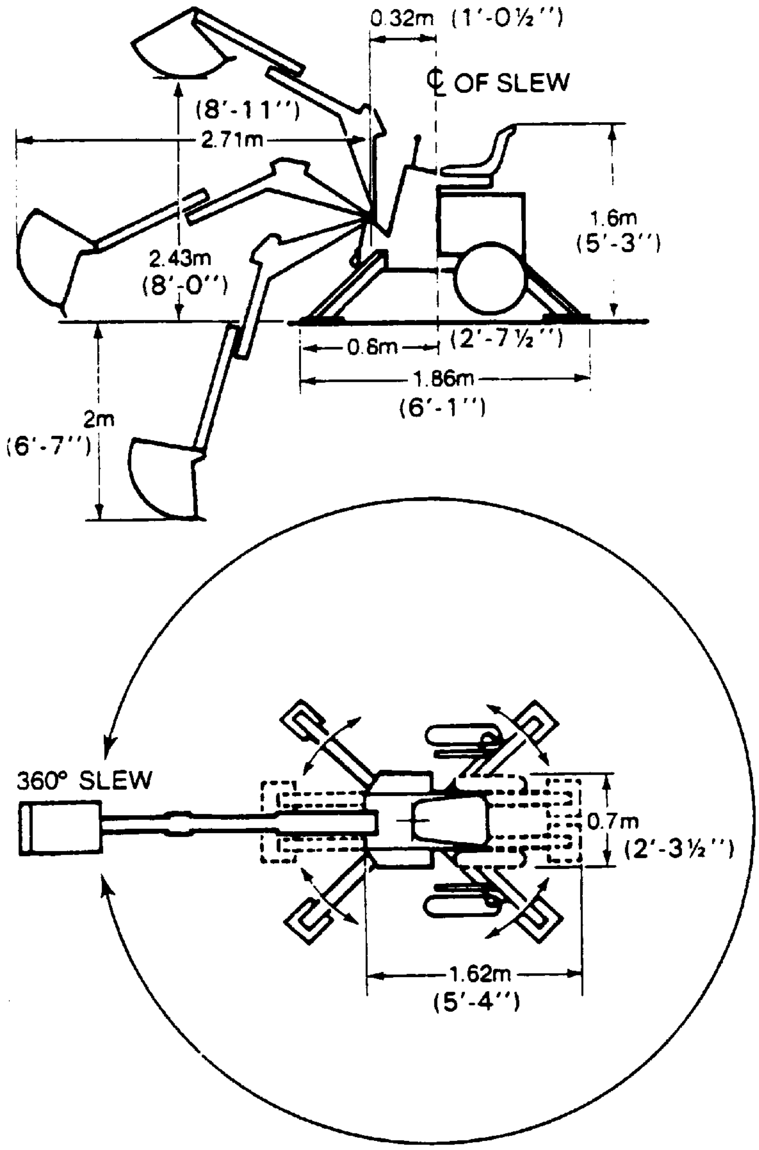


FIGURE 2.5

Position of the Outriggers (POWERFAB[10])

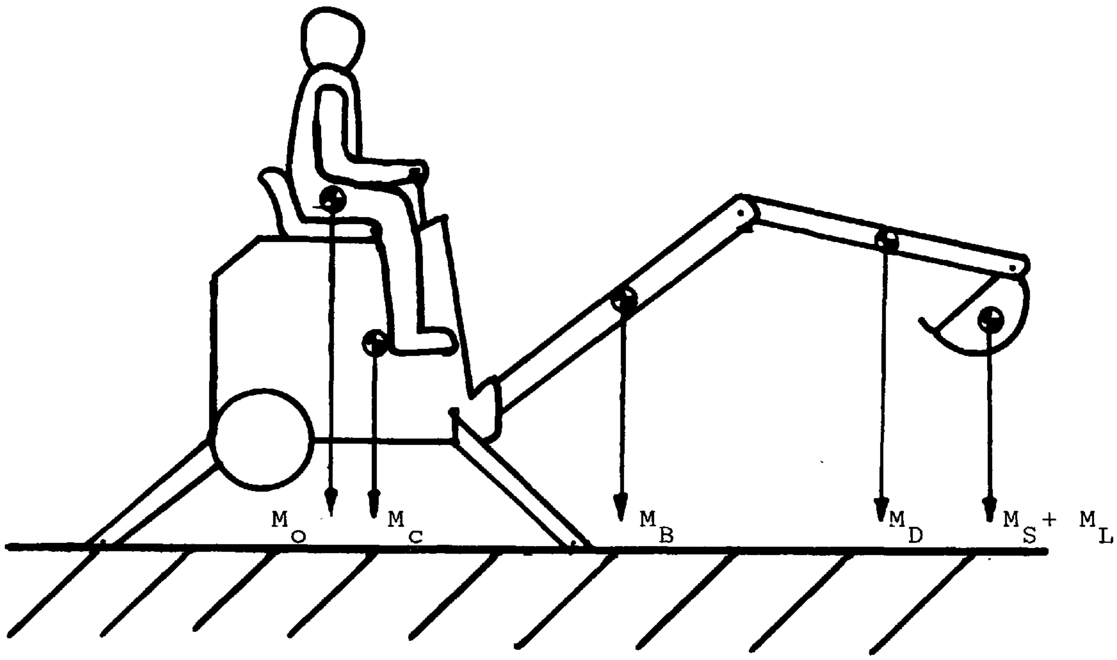


FIGURE 2.6

Static Stability While Lifting a Load

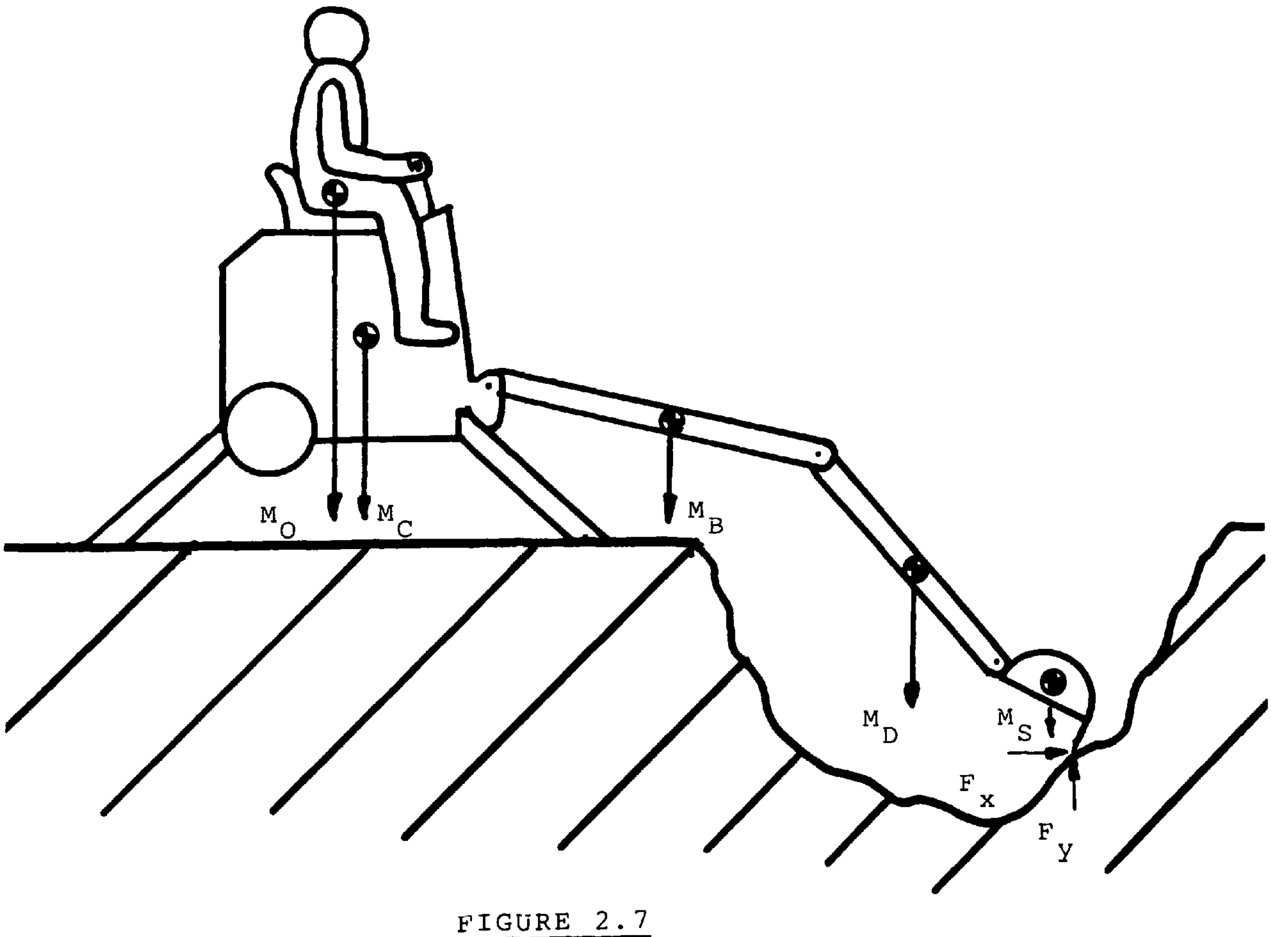


FIGURE 2.7

Static Stability While Digging the Ground

2.3.2 Static Stability While Digging

When the operator is using the microexcavator in a typical digging situation the bucket tooth forces applied using the hydraulic cylinders come into contact with the ground at a variety of angles (Figure 2.7). It is possible to raise the front or rear outriggers off the ground whilst applying digging forces at various points in the operation. The limiting stability condition again acts as a 'safety' limit. The achievable digging force is therefore a function of the position of the digging arm in space and the available hydraulic cylinder forces.

2.3.3. Static Stability While the Lower Platform is Fixed

When the lower platform is fixed to the ground the microexcavator is 'infinitely' statically stable at all times, the achievable digging or lifting force is then only a function of the available hydraulic cylinder forces.

CHAPTER THREE

3. COMPUTER MODELLING THEORY

This chapter discusses the computer modelling theories developed for the structural modelling and the application to the design optimisation of the Powerfab 360 Microexcavator digging arm. It was decided to adopt a fundamental and logical approach to the problem using proven theories. The modelling theories were easily implemented on a low-cost microcomputer system.

3.1 Structural Modelling of the Digging Arm

During a typical digging cycle the microexcavator digging arm moves through a variety of geometric configurations with varying bucket tooth forces (Figure 3.1). Experience has shown that maximum bucket tooth forces are achieved when the bucket tooth is in contact with an immovable object when maximum force is applied using the bucket hydraulic cylinder. The hydraulic system is so designed to minimise the effects of dynamic forces on the structure hence maximum forces and stresses occur in essentially static of quasi-static conditions.

A simple structural model was set up to analyse the effects of the static forces. The effects out of plane bending, torsion, joint friction and lack of fit were ignored to simplify the problem. The structure was also assumed to be of negligible weight.

The digging arm was modelled as a number of simple rigid links as illustrated in Figure 3.2. The geometric configuration of the digging arm is dependent on the hydraulic cylinder lengths. For known linkage and hydraulic cylinder dimensions the geometric angles of the joints can be found by trigonometry as follows:-

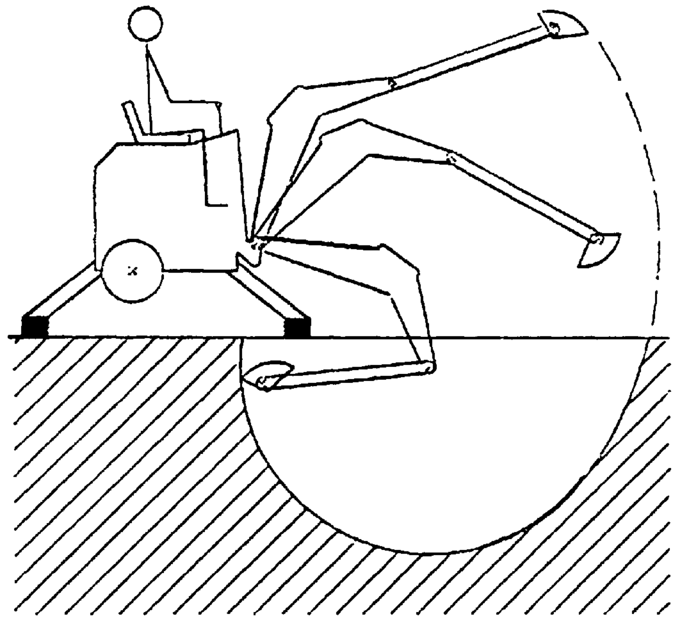


FIGURE 3.1

Geometric Configurations During a Typical Digging Cycle

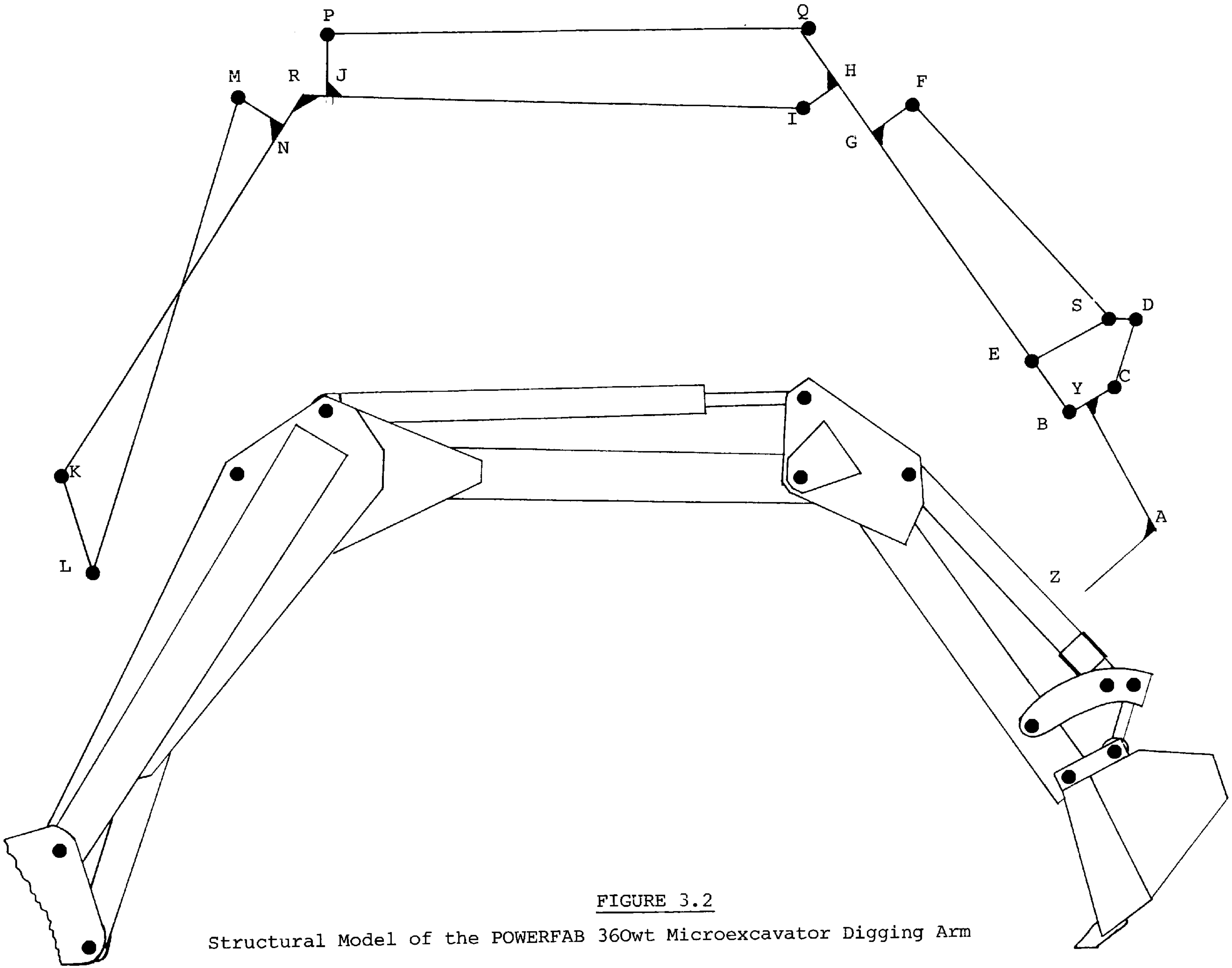


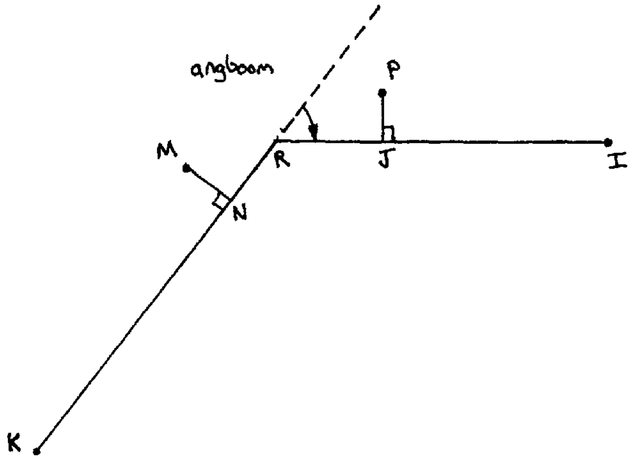
FIGURE 3.2

Structural Model of the POWERFAB 360wt Microexcavator Digging Arm

3.1.1 Calculation of Geometric Angles

In all calculations the valid range of angles is assumed to be 0–180° unless otherwise stated. The Cosine rule is used wherever necessary since a unique value of Cos of an angle exists over the range 0–180°

For the Boom Arm Geometry:-

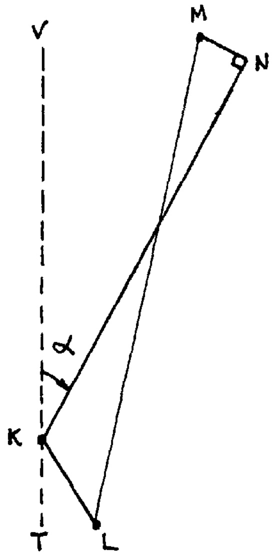


For triangle IKR

$$I\hat{R}K = \cos^{-1}\left[\frac{l_{KR}^2 + l_{IR}^2 - l_{IK}^2}{2\, l_{KR}\, l_{IR}}\right]$$

$$\text{angboom} = \pi - I\hat{R}K \qquad (3.1)$$

For the lower Boom Geometry:-



For triangle KMN

$$l_{KM} = \sqrt{l_{KN}^2 + l_{MN}^2}$$

$$K\hat{M}N = \cos^{-1}\left[\frac{l_{KM}^2 + l_{KL}^2 - l_{KN}^2}{2\, l_{KN}\, l_{KL}}\right]$$

$$M\hat{K}N = \frac{\pi}{2} - K\hat{M}N$$

For triangle KLM

$$L\hat{K}M = \cos^{-1}\left[\frac{l_{KM}^2 + l_{KL}^2 - l_{LM}^2}{2\, l_{KM}\, l_{KL}}\right]$$

$$L\hat{K}N = L\hat{K}M - M\hat{K}N$$

$$K\hat{M}L = \cos^{-1}\left[\frac{l_{KM}^2 + l_{LM}^2 - l_{KL}^2}{2\, l_{KM}\, l_{LM}}\right]$$

$$L\hat{M}N = K\hat{M}N - K\hat{M}L$$

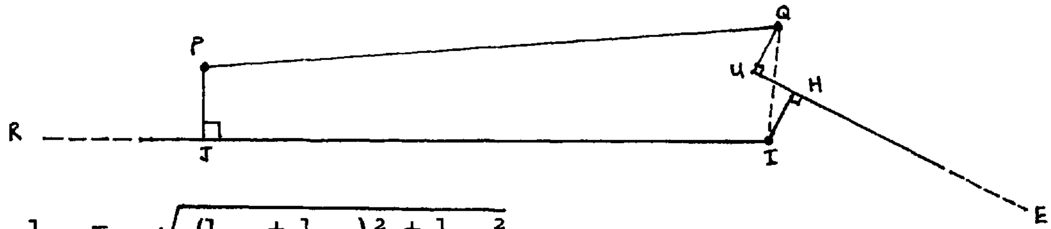
$$K\hat{L}M = \cos^{-1}\left[\frac{l_{KL}^2 + l_{LM}^2 - l_{KM}^2}{2\, l_{KL}\, l_{LM}}\right]$$

The angle of the lower boom to the vertical $N\hat{K}V$ is calculated as follows:-

$$L\hat{K}T = \mathrm{TAN}^{-1}\left[\frac{l_{LT}}{l_{KT}}\right]$$

$$N\hat{K}V = \pi - (L\hat{K}T + L\hat{K}N) \qquad (3.2)$$

For the upper boom geometry:-



$$l_{IQ} = \sqrt{(l_{HI} + l_{QU})^2 + l_{HM}^2}$$

$$l_{IP} = \sqrt{l_{JP}^2 + l_{IJ}^2}$$

For triangle IJP

$$I\hat{P}J = \mathrm{TAN}^{-1}\left[\frac{l_{IJ}}{l_{JP}}\right]$$

and

$$J\hat{I}P = \frac{\pi}{2} - I\hat{P}J$$

$$I\hat{P}Q = \cos^{-1}\left[\frac{l_{IP}^2 + l_{PQ}^2 - l_{IQ}^2}{2\, l_{IP}\, l_{PQ}}\right]$$

$$J\hat{P}Q = I\hat{P}J + I\hat{P}Q \qquad (3.3)$$

For triangle HIQ

$$H\hat{I}Q = \mathrm{TAN}^{-1}\left[\frac{l_{HU}}{l_{HI} + l_{QU}}\right]$$

$$P\hat{I}Q = \mathrm{COS}^{-1}\left[\frac{l_{IP}^{2} + l_{IQ}^{2} - l_{PQ}^{2}}{2\, l_{IP}\, l_{IQ}}\right]$$

$$H\hat{I}J = H\hat{I}Q + P\hat{I}Q + J\hat{I}P \qquad (3.4)$$

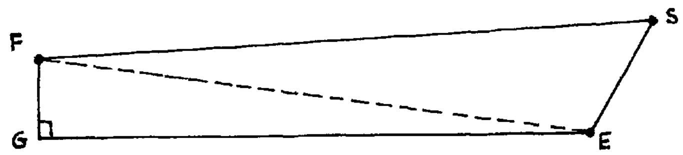
$$I\hat{Q}U = H\hat{I}Q$$

$$I\hat{Q}P = H\hat{I}Q$$

$$I\hat{Q}P \quad \mathrm{COS}^{-1}\left[\frac{l_{PQ}^{2} + l_{IQ}^{2} - l_{IP}^{2}}{2\, l_{PQ}\, l_{IQ}}\right]$$

$$P\hat{Q}U = I\hat{Q}P - I\hat{Q}U \qquad (3.5)$$

For the dipper arm geometry:-



$$l_{EF} = \sqrt{l_{FG}^{2} + l_{EG}^{2}}$$

$$F\hat{E}F = \mathrm{TAN}^{-1}\left[\frac{l_{FG}}{l_{EG}}\right]$$

$$E\hat{F}G = \frac{\pi}{2} - F\hat{E}G$$

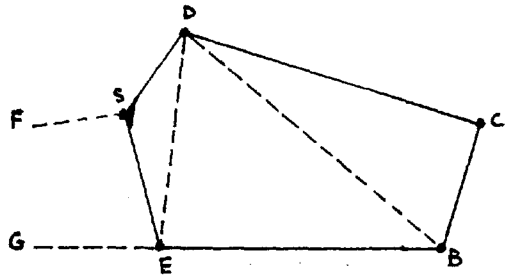
$$E\hat{S}F = \mathrm{COS}^{-1}\left[\frac{l_{FS}^{2} + l_{ES}^{2} - l_{EF}^{2}}{2\, l_{FS}\, l_{ES}}\right] \qquad (3.6)$$

$$E\hat{F}S = \mathrm{COS}^{-1}\left[\frac{l_{FS}^{2} + l_{EF}^{2} - l_{ES}^{2}}{2\, l_{FS}\, l_{EF}}\right]$$

$$G\hat{F}S = E\hat{F}S + E\hat{F}G \qquad (3.7)$$

$$F\hat{E}S = \pi - (E\hat{F}S + E\hat{S}F)$$

$$G\hat{E}S = F\hat{E}G + F\hat{E}S \qquad (3.8)$$



$$D\hat{E}S = \cos^{-1}\left[\frac{l_{ES}^2 + l_{DE}^2 - l_{DS}}{2\, l_{ES}\, l_{DE}}\right]$$

$$E\hat{D}S = \cos^{-1}\left[\frac{l_{DS}^2 + l_{DE}^2 - l_{ES}^2}{2\, l_{DS}\, l_{DE}}\right]$$

$$D\hat{S}E = \pi - (D\hat{E}S + E\hat{D}S)$$

$$B\hat{E}D = \pi - (G\hat{E}S + D\hat{E}S)$$

$$B\hat{E}S = B\hat{E}D + D\hat{E}S \qquad (3.9)$$

$$l_{BD} = \sqrt{l_{BE}^2 + l_{DE}^2 - 2.l_{BE}.l_{DE}.\cos\,(B\hat{E}D)}$$

$$B\hat{C}D = \cos^{-1}\left[\frac{l_{BC}^2 + l_{CD}^2 - l_{BD}^2}{2\, l_{BC}\, l_{CD}}\right]$$

$$B\hat{D}E = \cos^{-1}\left[\frac{l_{BD}^2 + l_{DE}^2 - l_{BE}^2}{2\, l_{BD}\, l_{DE}}\right]$$

$$D\hat{B}E = \cos^{-1}\left[\frac{l_{BD}^2 + l_{BE}^2 - l_{DE}^2}{2\, l_{BD}\, l_{BE}}\right]$$

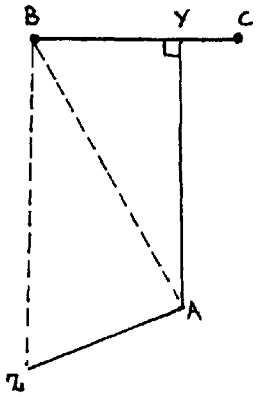
$$C\hat{B}D = \cos^{-1}\left[\frac{l_{BD}^2 + l_{BC}^2 - l_{CD}^2}{2\, l_{BD}\, l_{BC}}\right]$$

$$B\hat{D}C = \cos^{-1}\left[\frac{l_{BD}^2 + l_{CD}^2 - l_{BC}^2}{2\, l_{BD}\, l_{CD}}\right]$$

$$C\hat{D}E = B\hat{D}C + B\hat{D}E$$

$$D\hat{B}E = C\hat{B}D + D\hat{B}E$$

For the bucket geometry:-



for ABY

$$l_{AB} = \sqrt{l_{AY}^2 + l_{BY}^2}$$

$$A\hat{B}Y = TAN^{-1}\left[\frac{l_{AY}}{l_{BY}}\right]$$

$$B\hat{A}Y = \pi - A\hat{B}Y$$

For triangle AZB

$$A\hat{Z}B = COS^{-1}\left[\frac{l_{AZ}^2 + l_{BZ}^2 - l_{AB}^2}{2\, l_{AZ}\, l_{BZ}}\right]$$

$$B\hat{A}Z = COS^{-1}\left[\frac{l_{AZ}^2 + l_{AB}^2 - l_{BZ}^2}{2\, l_{AZ}\, l_{AB}}\right]$$

$$A\hat{B}Z = \pi - (A\hat{Z}B + B\hat{A}Z)$$

$$Y\hat{B}Z = A\hat{B}Y + A\hat{B}Z \qquad (3.10)$$

$$Y\hat{A}Z = B\hat{A}Y + B\hat{A}Z \qquad (3.11)$$

3.1.2 Calculation of the Position of the major Joints in Space

Before any calculation of forces on the structural model can be made, the X-Y positions of the major joints in space must be determined; this is particularly important for the calculation of the forces acting on the chasis due to a given tooth force at Z:

Figure 3.3 illustrates the four major 'links' that constitute the digging and these may be used to calculate the X-Y positions of major joints K, R, I, B and Z.

The angles between the major links at the major joints may be calculated as follows. (These are all assumed to lie in the range 0-180 degrees):-

$\underline{\alpha}$

Angle α can be found from equation (3.2) since:-

$$\alpha = \hat{NKV} \tag{3.12}$$

<u>angboom</u>

Angle angboom can be found equation (3.1)

$\underline{\beta}$

Angle β can be found as follows:-

$$l_{BI} = \sqrt{l_{BH}^2 + l_{HI}^2}$$

$$\hat{BIH} = TAN^{-1}\left[\frac{l_{BH}}{l_{HI}}\right]$$

$$\beta = (\hat{HIJ} + \hat{BIH}) - \pi \tag{3.13}$$

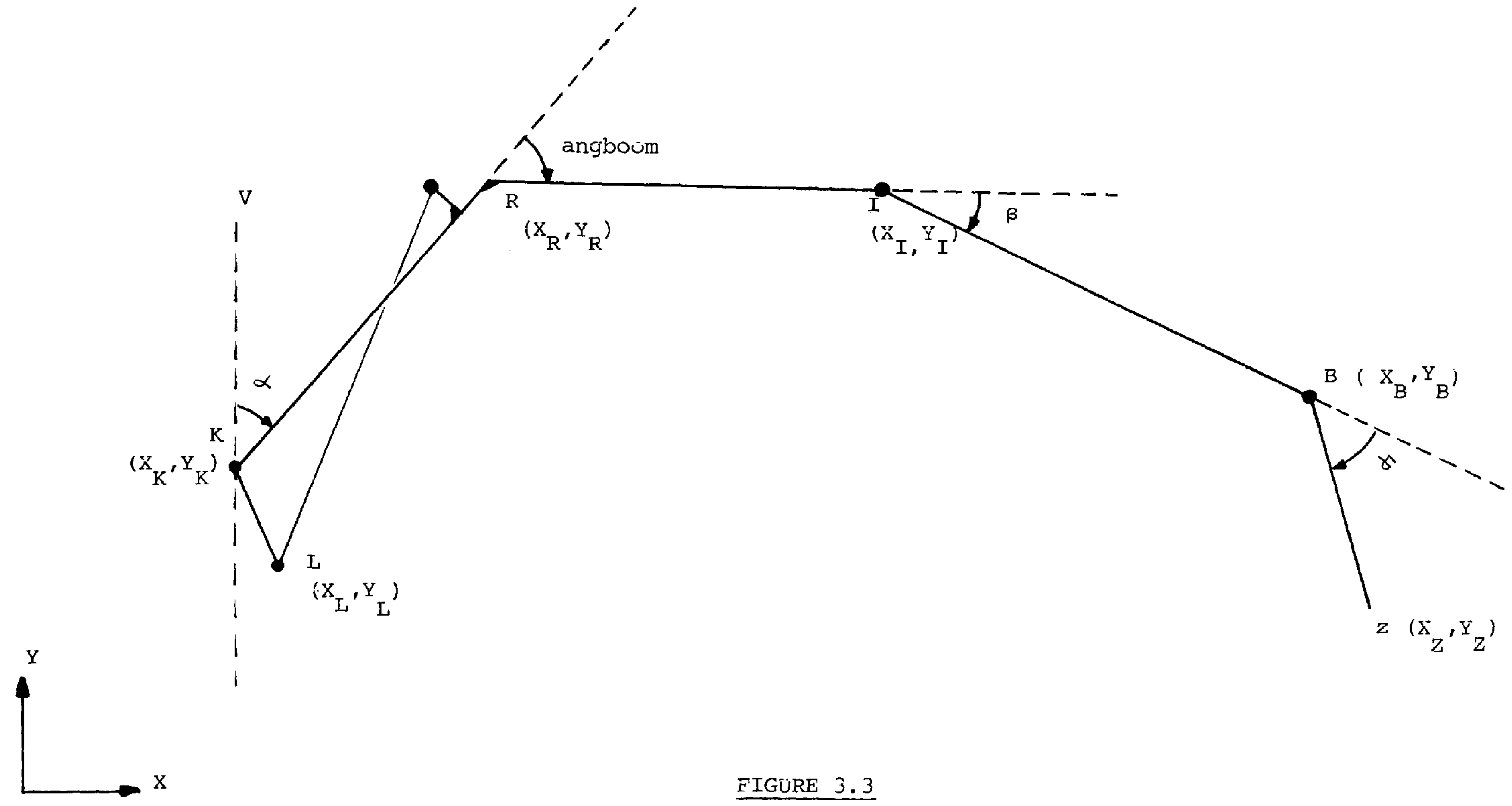


FIGURE 3.3

Position of the Major Joints of the Digging Arm in Space

$\underline{\gamma}$

Angle γ can be found as follows:-

$$\hat{HBI} = \frac{\pi}{2} - \hat{BIH}$$

$$\gamma = \hat{CBE} + \hat{HBI} + \hat{YBZ} - \pi \qquad (3.14)$$

$\underline{\Delta}$

Angle Δ can be found as follows:-

$$\Delta = \pi - \hat{LKT} \qquad (3.15)$$

The X-Y positions of the major joints can now be determined:-

<u>Joint K</u>

This joint connects the lower boom to the upper platform of the microexcavator, it is used as a reference point for the calculation of joint co-ordinates, hence:

$$X_K = 0$$

$$Y_K = 0$$

<u>Joint L</u>

This joint connects the 'base' end of the boom cylinder to the upper platform, the X-Y co-ordinates are:-

$$X_L = X_K + l_{KL} \text{ SIN } (\Delta) \qquad (3.16)$$

$$Y_L = Y_K + l_{KL} \text{ COS } (\Delta) \qquad (3.17)$$

Joint R

This welded joint connects the upper and lower boom structure, the X-Y co-ordinates are:-

$$X_R = X_K + l_{KR}\,\mathrm{SIN}\ (\alpha) \tag{3.18}$$

$$Y_R = Y_K + l_{KR}\,\mathrm{COS}\ (\alpha) \tag{3.19}$$

Joint I

This joint connects the upper boom and dipper arm, the X-Y co-ordinates are:-

$$X_I = X_R + l_{IR}\,\mathrm{SIN}\ (\alpha + \mathrm{angboom}) \tag{3.20}$$

$$Y_I = Y_R + l_{IR}\,\mathrm{COS}\ (\alpha + \mathrm{angboom}) \tag{3.21}$$

Joint B

This joint connects the dipper arm and the bucket, the X-Y co-ordinates are:-

$$X_B = X_I + l_{BI}\,\mathrm{SIN}\ (\alpha + \mathrm{angboom} + \beta) \tag{3.22}$$
$$Y_B = Y_I + l_{BI}\,\mathrm{COS}\ (\alpha + \mathrm{angboom} + \beta) \tag{3.23}$$

Joint Z

This joint denotes the position of the tip of the bucket tooth, the X-Y co-ordinates are:-

$$X_Z = X_B + l_{BZ}\,\mathrm{SIN}\ (\alpha + \mathrm{angboom} + \beta + \gamma) \tag{3.24}$$

$$Y_Z = Y_B + l_{BZ}\,\mathrm{COS}\ (\alpha + \mathrm{angboom} + \beta + \gamma) \tag{3.25}$$

3.1.3 Calculation of Forces at the Joints

Once the geometric angles of the structural model and the X-Y position of the joints in space have been determined, the magnitude and direction of the forces at the joints can be found using fundamental statics.

Initially a suitable notation must be selected before any calculation can be made. Figure 3.4 shows the interaction of forces between the major component of the digging arm, for a given applied force at the bucket tooth joint Z.

The entire digging arm mechanism and each individual component must satisfy the following fundamental static equations as follows (Boothroyd[11]):-

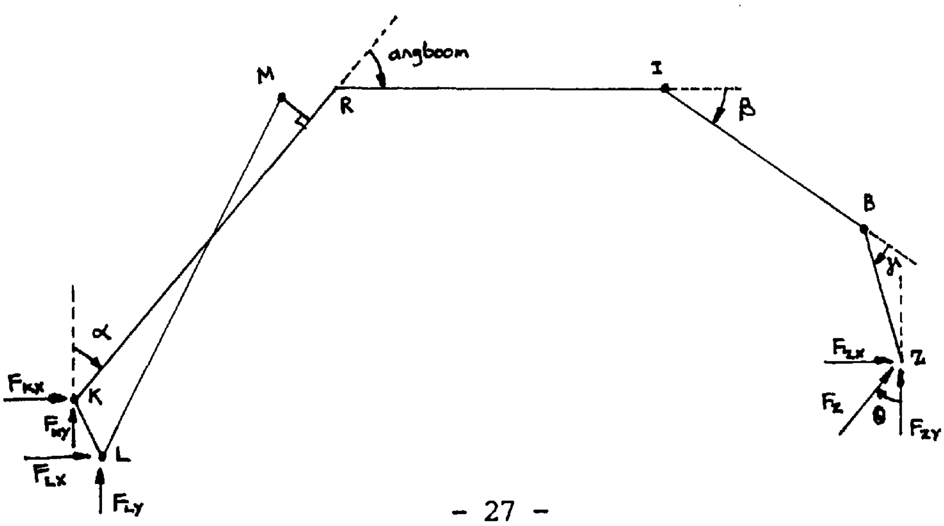
$$F_X = 0 \tag{3.26}$$

$$F_Y = 0 \tag{3.27}$$

$$M_P = 0 \tag{3.28}$$

Forces at joints K and L

For given linkage mechanism dimensions and an applied force at joint Z, the forces at the lower boom hinge K and boom cylinder L can be found as follows:-



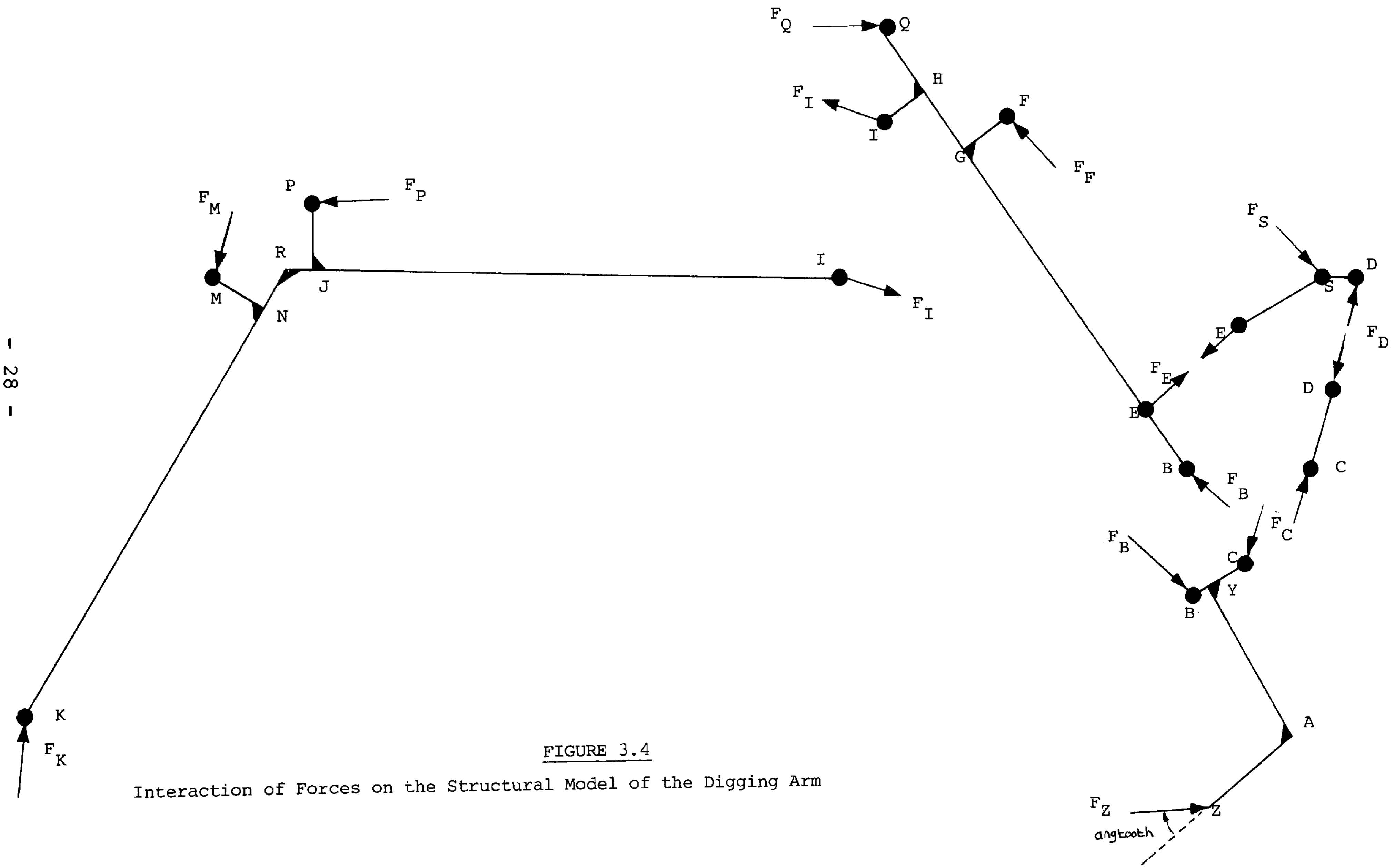


FIGURE 3.4

Interaction of Forces on the Structural Model of the Digging Arm

Resolving horizontally, from Equation 3.26:-

$$F_{ZX} + F_{KX} + F_{LX} = 0 \tag{3.29}$$

Resolving vertically, from Equation 3.27:-

$$F_{ZY} + F_{KY} + F_{LY} = 0 \tag{3.30}$$

The angle of the applied force at Z to the vertical, θ can be found as follows:-

$$\theta = \alpha + \text{angboom} + \beta + \gamma - (\pi - A\hat{Z}B) + \text{angtooth}$$

Resolving the force at Z:-

$$F_{ZX} = F_Z \text{ SIN } (\theta) \tag{3.31}$$

$$F_{ZY} + F_Z \text{ COS } (\theta) \tag{3.32}$$

Taking moments about joint K for convenience and using equation 3.28 and +VE clockwise moments :-

$$-1 \; (F_{XZ} \; (Y_Z - Y_K) + F_{ZY} \; (X_Z - X_K) + F_{LX} \; (Y_L - Y_K) + F_{LY} \; (X_L - X_K)) = 0 \tag{3.33}$$

Link 1_{IM} is a two-pinned hydraulic cylinder therefore the forces at joint L must act along the line of the hydraulic cylinder in the direction L-M.

$\therefore$ Resolving forces at joint L:-

$$\text{TAN } \psi = \frac{F_{LY}}{F_{LX}}$$

$$F_{LY} = \text{TAN}(\psi) \; F_{LX} \tag{3.34}$$

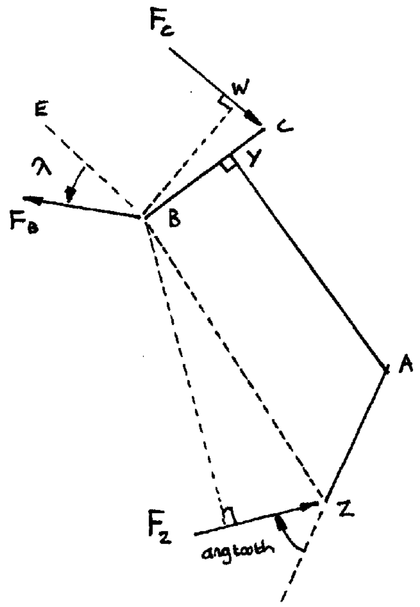
Substituting in equation 3.33 and rearranging

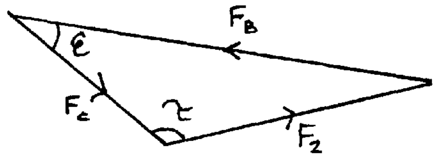
$$F_{LX} = \left[\frac{F_{ZX} \ (Y_Z - Y_K) - F_{ZY} \ (X_Z - X_K)}{\mathrm{TAN}\ \psi \ (X_L - X_K) + (Y_L - Y_K)} \right] \qquad (3.35)$$

And F_{LY} can be determined from equation 3.34

Having determined forces F_{ZX}, F_{ZY}, F_{LX}, F_{LY}
F_{KZ} can be found from equation 3.29 and
F_{KY} can be found from equation 3.30

<u>Forces at Joints B and C</u>





For a given applied force at joint Z the forces at joints B and C can be found using the triangle of forces:-

$$l_{BW} = l_{BC} \ \mathrm{SIN}\ \hat{BCD}$$

Taking movements about joint B for convenience +VE clockwise:-

$$F_C \, l_{BW} = F_Z \, l_{BZ} \, \mathrm{SIN} \, (\pi - \hat{AZB} - \mathrm{angtooth})$$

hence

$$F_C = \frac{l_{BC} \, \mathrm{SIN} \, (\pi - \hat{AZB} - \mathrm{angtooth})}{l_{BW}} F_Z \qquad (3.36)$$

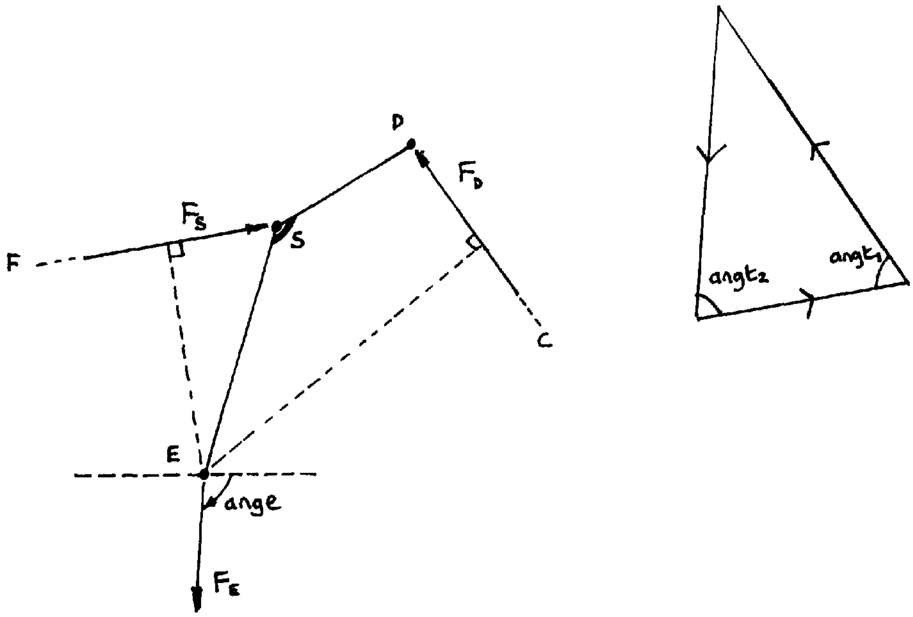
$$\Upsilon = \frac{3\pi}{2} + \mathrm{angtooth} - (\hat{BCD} + \hat{YAZ})$$

$$F_B = \sqrt{F_C^2 + F_Z^2 - 2 \, F_C \, F_Z \, \mathrm{COS} \, (\Upsilon)}$$

$$\epsilon = \mathrm{COS}^{-1} \left[\frac{F_B^2 + F_C^2 - F_Z^2}{2 \, F_B \, F_C} \right]$$

$$\lambda = \epsilon + \pi - (\hat{BCD} + \hat{CBE})$$

Forces at joints E and S



Links FS and C-D are two pin links so the transmitted force between the pins must act along the line of the links such that $F_D = F_C$ and $F_F = F_S$ and F_C is known from equation 3.36.

Taking moments about joint E for convenience, clockwise +VE:-

$$F_S\ l_{ES}\ \mathrm{SIN}\ E\hat{S}F = F_D\ l_{DE}\ \mathrm{SIN}\ C\hat{D}E$$

so $$F_S = \frac{l_{DE}\ \mathrm{SIN}\ C\hat{D}E\ F_D}{l_{ES}\ \mathrm{SIN}\ E\hat{S}F} \tag{3.37}$$

$$C\hat{D}S = C\hat{D}E + E\hat{D}S$$

$$\mathrm{angt1} = 2\pi - (E\hat{S}F + D\hat{S}E + C\hat{D}S)$$

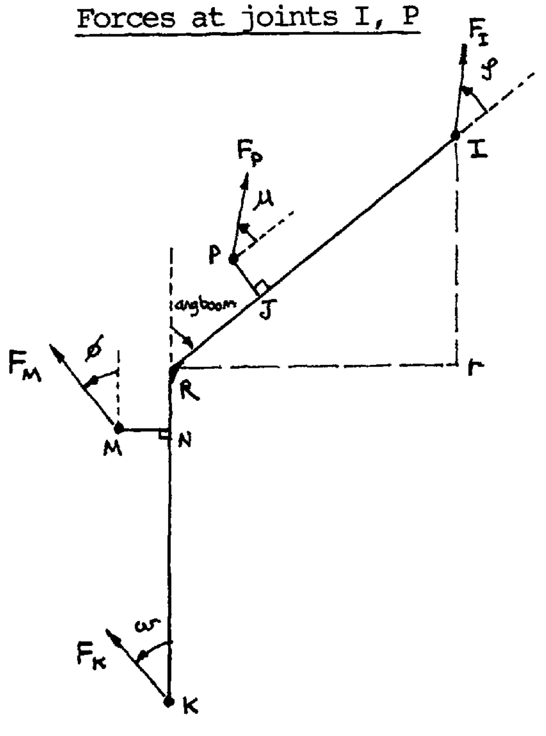
$$F_E = \sqrt{F_D{}^2 + F_S{}^2 - 2\ F_D\ F_S\ \mathrm{COS}\ (\mathrm{angt1})} \tag{3.38}$$

$$\mathrm{angt2} = \mathrm{COS}^{-1}\left[\frac{F_E{}^2 + F_S{}^2 - F_D{}^2}{2\ F_E\ F_S}\right]$$

$$\mathrm{angt3} = B\hat{E}S - E\hat{S}F$$

$$\mathrm{ange} = \pi - (\mathrm{angt2} + \mathrm{angt3})$$

<u>Forces at joints I, P</u>



Links L-M and P-Q are hydraulic cylinders so the transmitted forces act along the links such that

$F_M = F_L$ and

$F_P = F_Q$

and F_L and F_K are found as follows:-

$$F_L = \sqrt{F_{LX}{}^2 + F_{LY}{}^2} - \tag{3.39}$$

$$F_K = \sqrt{F_{KX}{}^2 + F_{KY}{}^2} - \tag{3.40}$$

The angles ω, ϕ, μ are calculated as follows:-

$$\mu = J\hat{P}Q - \frac{\pi}{2}$$

$$\phi = \frac{\pi}{2} - L\hat{M}N$$

$$\omega = \psi + N\hat{K}V - \frac{\pi}{2}$$

To simplify the problem of calculating the forces, the forces at the joints are resolved into horizontal and vertical components.

For joint K

$$F_{KH} = F_K \text{ SIN } (\omega)$$

$$F_{KV} = F_K \text{ COS } (\omega)$$

For joint M

$$F_{MH} = F_M \text{ SIN } (\phi)$$

$$F_{MV} = F_M \text{ COS } (\phi)$$

For force balance from equation 3.26

$$F_{KH} + F_{MH} + F_{PH} + F_{IH} = 0 \qquad (3.41)$$

and from equation 3.27

$$F_{KV} + F_{MV} + F_{PV} + F_{IV} = 0 \qquad (3.42)$$

Taking movements about joint I for convenience +VE clockwise:-

$$F_{KV}\, l_{Rr} + F_{KH}\,(l_{Ir} + l_{KR}) + F_{MVY}\,(l_{Rr} + l_{MN})$$

$$+ F_{MH}\,(l_{Ir} + l_{NR}) + F_{PH}\, l_{JP} + F_{PV}\, l_{IJ} = 0 \qquad (3.43)$$

but $F_{PH} = F_P \text{ SIN (angboom } -\mu)$

$F_{PV} = F_P \text{ COS (angboom } -\mu)$

Substituting in equation 3.43 and re-arranging:-

$$F_P = -1\left[\frac{F_{KV}l_{Rr}+F_{KH}(l_{Ir}+l_{KR})+F_{MV}\ (l_{Rr}+l_{MN}) + F_{MH}\ (l_{Ir}+l_{NR})}{(l_{JP} + l_{IJ})}\right] \tag{3.44}$$

Hence F_{PH} and F_{PV} can be found as follows:-

$$F_{PH} = F_P \cos(\mu)$$
$$F_{PV} = F_P \sin(\mu)$$

F_{IH} and F_{IV} can be found from equations 3.41 and 3.42 as follows:-

$$F_{IH} = -1\ (F_{KH} + F_{MH} + F_{PH}) \tag{3.45}$$

$$F_{IV} = -1\ (F_{KV} + F_{MV} + F_{PV}) \tag{3.46}$$

and angle $\mathcal{J}$ is given by:-

$$\mathcal{J} = \text{angboom} - \text{TAN}^{-1}\left[\frac{F_{IH}}{F_{IV}}\right] \tag{3.47}$$

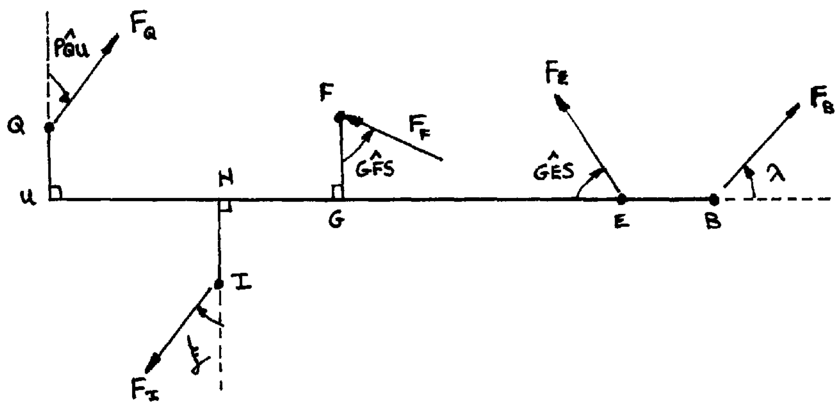
<u>Forces at joints Q and F</u>

Links P-Q and F-S are two pinned (hydraulic cylinder), so the transmitted force again acts along the length of each link only. The forces at joints Q and F are therefore:-

$$F_F = F_S \tag{3.48}$$

$$F_Q = F_P \tag{3.49}$$

Having calculated all of the forces at the joints on the structural model a force and moment balance is carried out on the dipper arm only, to test for the errors in calculations:-



The angles $\hat{PQU}$, $\hat{IQU}$ AND $\hat{IQP}$ are found as follows:-

$$\hat{IQP} = \cos^{-1}\left[\frac{l_{PQ}^2 + l_{IQ}^2 - l_{IP}^2}{2\, l_{PQ}\, l_{IQ}}\right]$$

$$\hat{IQU} = \hat{HIQ}$$

$$\hat{PQU} = \hat{IQP} - \hat{IQU}$$

Resolving horizontally and from equation 3.26:-

$$F_Q \mathrm{SIN}(\hat{PQU}) + F_B \mathrm{COS}(\lambda) - F_I \mathrm{SIN}(\gamma) - F_F \mathrm{SIN}(\hat{GFS}) = 0$$

Resolving vertically and from equation 3.27:-

$$F_Q \mathrm{COS}(\hat{PQU} + F_B \mathrm{SIN}(\lambda) + F_F \mathrm{COS}(\hat{GFS}) - F_I \mathrm{COS}(\gamma) = 0$$

Taking moments about joint B for convenience K +VE clockwise and using equation 3.28 :-

$$F_E\ \mathrm{SIN}\ (\hat{GES})\ l_{BE} + F_F\ \mathrm{COS}\ (\hat{GFS})\ l_{BG} + F_I\ \mathrm{SIN}\ (\gamma)\ l_{HI}$$

$$+ F_Q\ \mathrm{COS}\ (\hat{PQU})\ l_{BU} + F_Q\ \mathrm{SIN}\ (\hat{PQU})\ l_{QU} - F_F\ \mathrm{SIN}\ (\hat{GFS})\ l_{FG}$$

$$- F_I\ \mathrm{COS}\ (\gamma)\ l_{BH} = 0$$

3.1.4 Calculation of Stresses at Locations of Interest on the Structural Model

Elementary structural mechanics theory can be used to calculate the stresses at various points on the major components of the structural model (Case and Chilvers[12]).

The direct stress σ_d at a section A_{xx} due to a direct load F_x is given by:-

$$\sigma_d = \frac{F_x}{A_{xx}} \tag{3.50}$$

The bending stress σ_b at a section A_{xx} due to a bending load F_Y is given by:-

$$\sigma_b = \frac{M_x Y}{I_{xx}} \tag{3.51}$$

The combined stress σ_c due to both direct load and bending load is given by:-

$$\sigma_c = \sigma_d + \sigma_b \tag{3.52}$$

3.1.4.1 Boom Arm

For ease of calculation of the stresses, this component is divided into two parts, the upper and lower boom.

Upper Boom Section

Figure 3.5(a) illustrates the forces acting on the upper boom. The forces at the joints are resolved to give horizontal and vertical forces for ease of calculation.

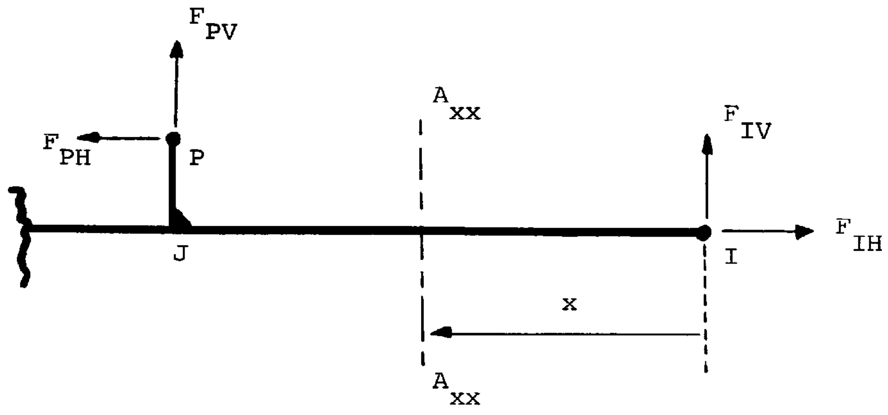


FIGURE 3.5(a)

Upper Boom

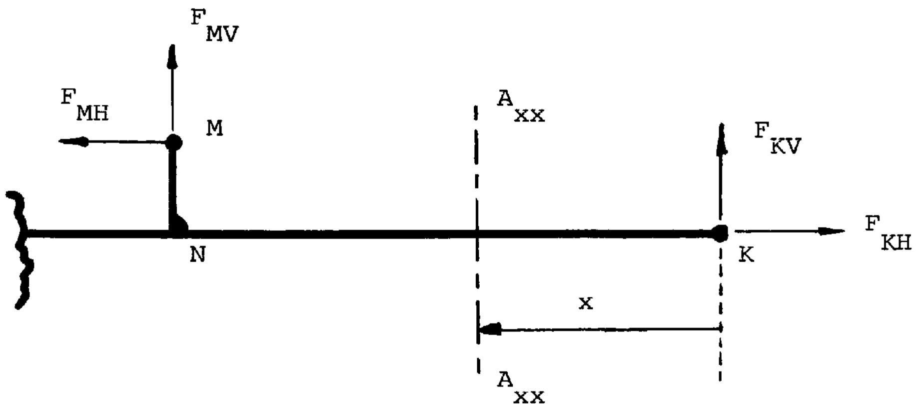


FIGURE 3.5(b)

Lower Boom

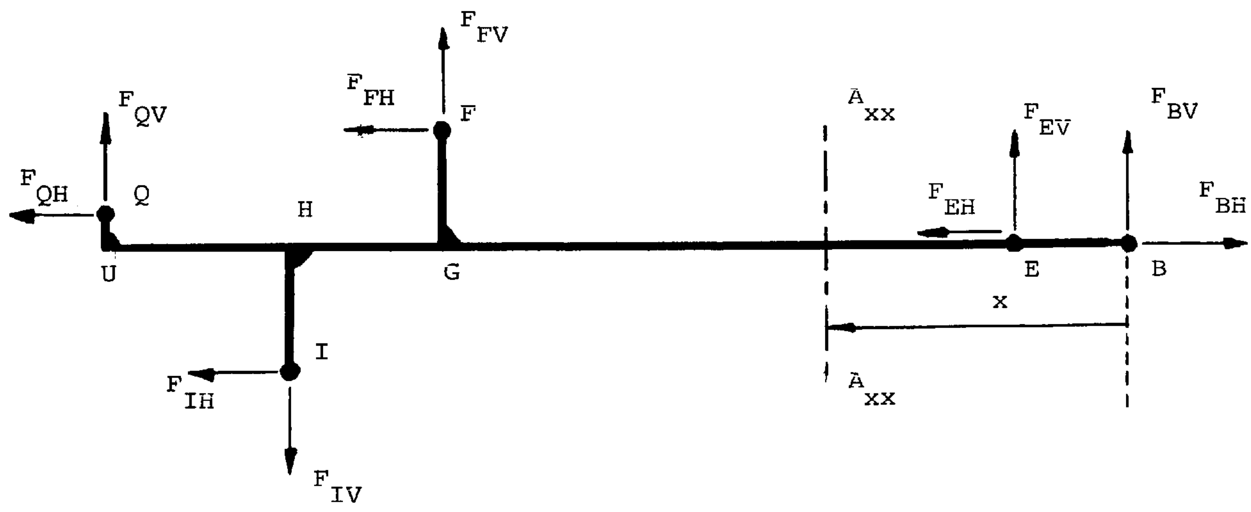


FIGURE 3.5(c)

Dipper Arm

FIGURE 3.5

Notation for the Forces Acting on the Structure

$\underline{0 \leq x < l_{IJ}}$

The direct stress σ_d at section A_{xx} is given by equation 3.50:-

$$\sigma_d = \frac{F_{IH}}{A_{xx}} \tag{3.53}$$

The bending stress σ_b at section A_{xx} is given by equation 3.51:-

$$\sigma_b = \frac{M_x \cdot y}{I_{xx}} \quad \frac{F_{IH} xy}{I_{xx}} \tag{3.54}$$

The combined stress σ_c at section A_{xx} is given by equation 3.52:-

$$\sigma_c = \sigma_d + \sigma_b$$

$\underline{l_{IJ} \leq x < l_{IR}}$

The direct stress σ_d at section A_{xx} is given by equation 3.50:-

$$\sigma_d = \frac{F_{IH} - F_{PH}}{A_{xx}} \tag{3.55}$$

The bending stress σ_b at section A_{xx} is given by equation 3.51:-

$$\sigma_b = \frac{M_x \cdot y}{I_{xx}} = \frac{(F_{IV} x + F_{PV}(x - l_{IJ}) + F_{PH} \cdot l_{JP}) \cdot y}{I_{xx}} \tag{3.56}$$

The combined stress σ_c at section A_{xx} is given by equation 3.52:-

Lower Boom Section

Figure 3.5(b) illustrates the forces acting on the lower boom. The forces at the joints are again resolved into horizontal and vertical forces for ease of calculation.

$0 \leq x < l_{KN}$

The direct stress σ_d at section A_{xx} is given by equation 3.50:-

$$\sigma_d = \frac{F_{KH}}{A_{xx}} \qquad (3.57)$$

The bending stress σ_b at section A_{xx} is given by equation 3.51:-

$$\sigma_b = \frac{M_x.y}{I_{xx}} = \frac{F_{KV}.x.y}{I_{xx}} \qquad (3.58)$$

The combined stress σ_c at section A_{xx} is given by equation 3.52:-

$l_{KN} \leq x < l_{KR}$

The direct stress σ_d at section A_{xx} is given by equation 3.50:-

$$\sigma_d = \frac{F_{KH} - F_{MH}}{A_{xx}} \qquad (3.59)$$

The bending stress σ_b at section A_{xx} is given by equation 3.51:-

$$\sigma_b = \frac{M_x.y}{I_{xx}} = \frac{(F_{KV}.x+F_{MV}(x-l_{KN})+F_{MH}l_{MN}).y}{I_{xx}} \qquad (3.60)$$

The combined stress σ_c at section A_{xx} is given by equation 3.52.

Dipper Arm Section

Figure 3.5(c) illustrates the forces acting on the dipper arm. The forces at the joints are again resolved into horizontal and vertical forces for ease of calculation.

$0 \leq x < l_{BE}$

The direct stress σ_d at section A_{xx} is given by equation 3.50:-

$$\sigma_d = \frac{F_{BH}}{A_{xx}} \tag{3.61}$$

The bending stress σ_b at section A_{xx} is given by equation 3.51:-

$$\sigma_b = \frac{M_x . y}{I_{xx}} = \frac{F_{BV}.x.y}{I_{xx}} \tag{3.62}$$

The combined stress σ_c at section A_{xx} is given by equation 3.52:-

$l_{BE} \leq x < l_{BG}$

The direct stress σ_d at section A_{xx} is given by equation 3.50:-

$$\sigma_d = \frac{F_{BH} - F_{EH}}{A_{xx}} \tag{3.63}$$

The bending stress σ_b at section A_{xx} is given by equation 3.51:-

$$\sigma_b = \frac{M_x . y}{I_{xx}} = \frac{(F_{BV}.x+F_{EV}(x-l_{BE})\ )y}{I_{xx}} \tag{3.64}$$

The combined stress σ_c at section A_{xx} is given by equation 3.52:-

$\underline{l_{BG} \leq x < l_{BH}}$

The direct stress σ_d at section A_{xx} is given by equation 3.50:-

$$\sigma_d = \frac{F_{BH} - F_{EH} - F_{FH}}{A_{xx}} \tag{3.65}$$

The bending stress σ_b at section A_{xx} is given by equation 3.51:-

$$\sigma_b = \frac{M_x \cdot y}{I_{xx}}$$

$$= \frac{F_{BV} \cdot x + F_{EV}(x - l_{BE}) + F_{FV}(x - l_{BG}) + F_{FH} l_{FG} \cdot y}{I_{xx}} \tag{3.66}$$

The combined stress σ_c at section A_{xx} is given by equation 3.52:-

$\underline{l_{BH} \leq x < l_{BH}}$

The direct stress σ_d at section A_{xx} is given by equation 3.50:-

$$\sigma_d = \frac{F_{BH} - F_{EH} - F_{FH} - F_{IH}}{A_{xx}} \tag{3.67}$$

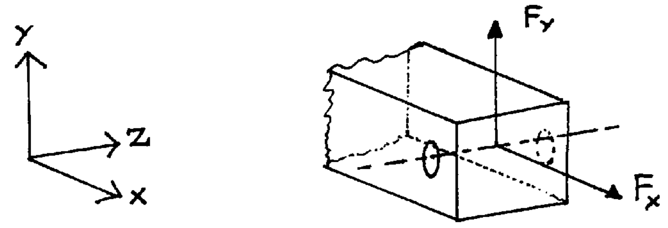
The bending stress σ_b at section A_{xx} is given by equation 3.51:-

$$\sigma_b = \frac{M_x \cdot y}{I_{xx}}$$

$$= \frac{\left[F_{BV} \cdot x + F_{EV}(x - l_{BE}) + F_{FV}(x - l_{BG}) + F_{FH} \cdot l_{FG} - F_{IV}(x - l_{BH}) - F_{IH} \cdot l_{HI}\right] \cdot y}{I_{xx}} \tag{3.68}$$

The combined stress σ_c at section A_{xx} is given by equation 3.52. equation.

3.1.5 Effects of Torsion on a hollow box section



Assumed forces on the upper boom section.

The various components of the microexcavator digging arm have been assumed to be loaded in one plane only, the X-Y plane as illustrated.

Generally for a two dimensional stress system the principal stresses are given by (Case and Chilvers[12]):-

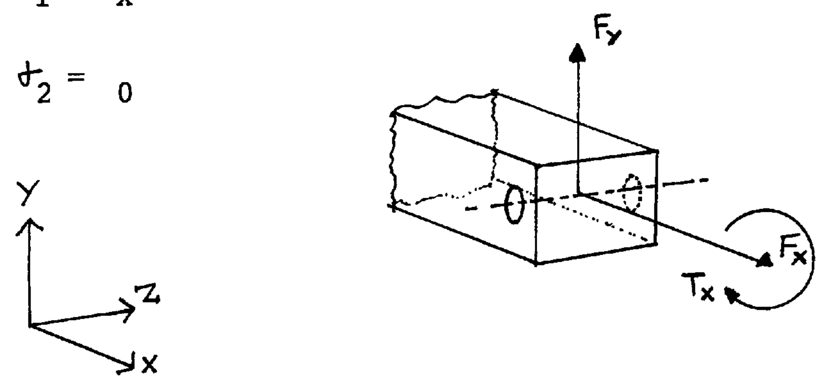
$$\sigma_1 = \frac{1}{2}(\sigma_x + \sigma_y) + \frac{1}{2}\sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2} \qquad (3.69)$$

$$\sigma_2 = \frac{1}{2}(\sigma_x + \sigma_y) - \frac{1}{2}\sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2} \qquad (3.70)$$

For loading in one plane only and ignoring the effects of vertical shear due to load F_y equations (3.69 and 3.70) become:-

$$\sigma_1 = \sigma_x \qquad (3.71)$$

$$\sigma_2 = 0 \qquad (3.72)$$



Introducing a torsional moment T_x gives rise to torsional stress in the Y-Z plane on the upper, lower and side sections of the box section. The equations for principal stresses (3.69 and 3.70) become:-

$$\sigma_1 = \frac{1}{2}\sigma_x + \frac{1}{2}\sqrt{\sigma_x^2 + 4\tau_{xy}^2} \qquad (3.73)$$

$$\sigma_2 = \frac{1}{2}\sigma_x - \frac{1}{2}\sqrt{\sigma_x^2 + 4\tau_{xy}^2} \qquad (3.74)$$

3.2 DESIGN OPTIMISATION

This chapter discusses the design optimisation parameters and design optimisation method. For a given structure and geometry the design parameter that can be readily adjusted is the relief valve pressure setting. This will in turn influence the bucket tooth force and consequently the safety factor at a given point on the structure. Optimisation of these three variables is discussed in the following sections.

3.2.1 Maximum Hydraulic Cylinder Forces

For any given geometric configuration the maximum achievable digging force at the bucket tooth is dependent upon the maximum hydraulic cylinder forces that can be achieved by each hydraulic cylinder. Each hydraulic cylinder will achieve a different bucket tooth force due to the service line relief valve pressure setting or the hydraulic supply pressure setting. The hydraulic cylinder may apply a force or may have a force induced upon it by one or more of the other hydraulic rams. The four loading conditions for the hydraulic cylinder are illustrated in Figure 3.6 ((a) (b) (c) (d)).

The maximum hydraulic cylinder force is also dependent upon the cylinder dimensions as illustrated in Figure 3.7.

When forces are applied by the hydraulic cylinder the maximum forces are determined by the hydraulic supply pressure for a given cylinder size. These forces may be either tensile (Figure 3.6(c) or compressive (Figure 3.6(d)).

Similarly when forces are induced on the hydraulic cylinder by the other cylinders the maximum forces are determined by the service line relief valve pressure for a given cylinder size. These forces may also be either tensile (Figure 3.6(a)) or compressive (Fig. 3.6(b)).

For each condition the maximum hydraulic cylinder force (applied or induced) may be determined from the following equation:-

$$F = P.A \qquad (3.75)$$

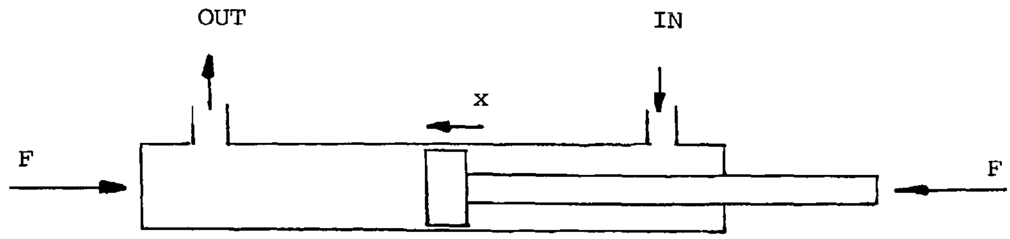


FIGURE 3.6(a)

Compressive Induced Forces

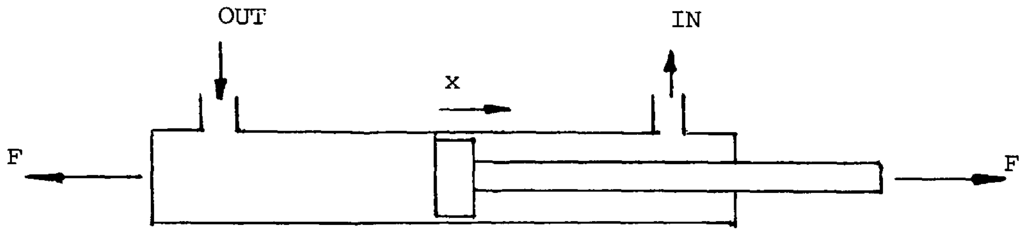


FIGURE 3.6(b)

Tensile Induced Forces

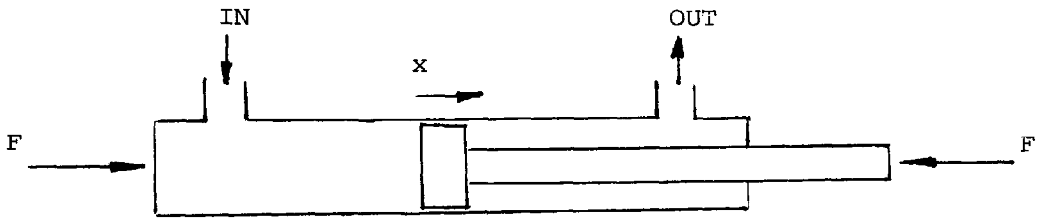


FIGURE 3.6(c)

Compressive Applied Forces

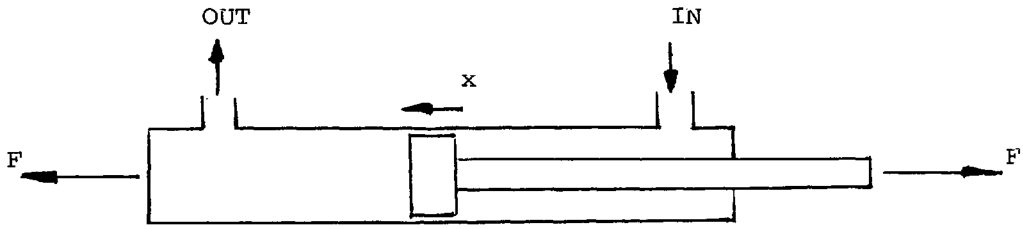


FIGURE 3.6(d)

Tensile Applied Forces

FIGURE 3.6

Hydraulic Cylinder Loading Conditions

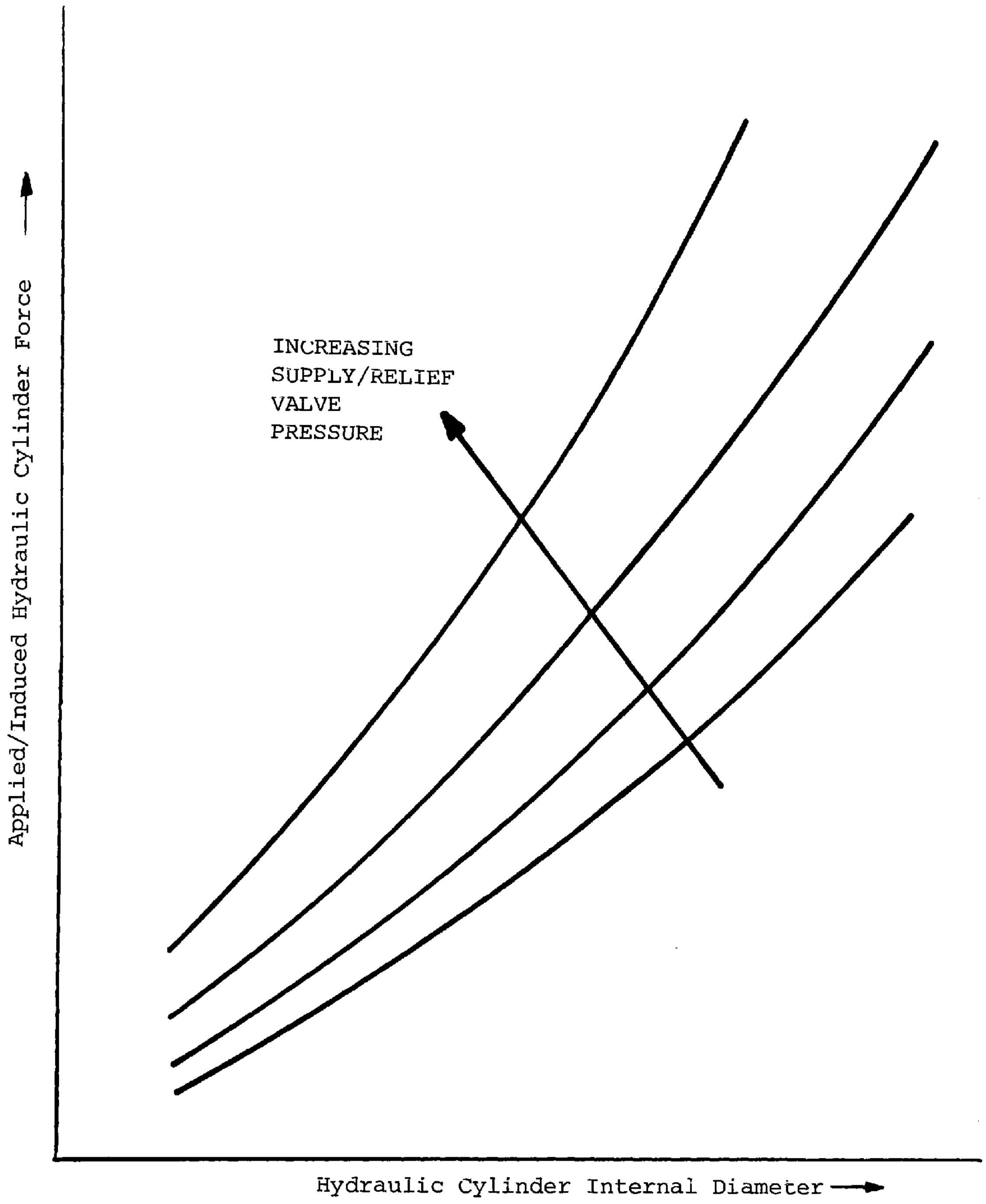


FIGURE 3.7

Variation of Applied/Induced Hydraulic Cylinder Force with Hydraulic Cylinder Diameter for Different Supply/Relief Valve Pressures

The maximum hydraulic cylinder forces are obtained under induced loading conditions when the pressure P is limited by the service line relief valve pressure setting which usually exceeds the supply pressure. Thus the maximum hydraulic cylinder forces are found in conditions Figure 3.6(a) and 3.6(b).

3.2.2 Determination of Maximum Digging Force

The maximum hydraulic cylinder forces can be determined as outlined in section 3.2.1, this section describes the method used to determine the maximum achievable digging force for a given geometric configuration.

Maximum bucket tooth forces are achieved when the bucket tooth forced is at right angles to the line drawn from the bucket hinge pin to the point of contact on the tip of the bucket tooth (Figure 3.8).

The fundamental structural model can be used to determine the maximum achievable digging force by applying a unit force at the bucket tooth for a known geometric configuration defined by the lengths of the three hydraulic cylinders. Using the simple structural model a tension coefficient for each hydraulic cylinder can be obtained due to the unit tooth force. For each hydraulic cylinder there will be a corresponding tension coefficient, which may be positive indicating a tensile force or negative indicating a compressive force.

The maximum bucket tooth force due to each hydraulic cylinder for a given geometric configuration is given by:-

For +VE tension coefficient,

Maximum bucket tooth force =

$$F_{tmax} = \frac{\text{Maximum induced tensile hydraulic cylinder force}}{\text{Hydraulic cylinder tension coefficient}} \qquad (3.76)$$

or

For -VE tension coefficient,

Maximum bucket tooth force =

$$F_{tmax} = \frac{\text{Maximum induced compressive hydraulic cylinder force}}{\text{Hydraulic cylinder tension coefficient}} \qquad (3.77)$$

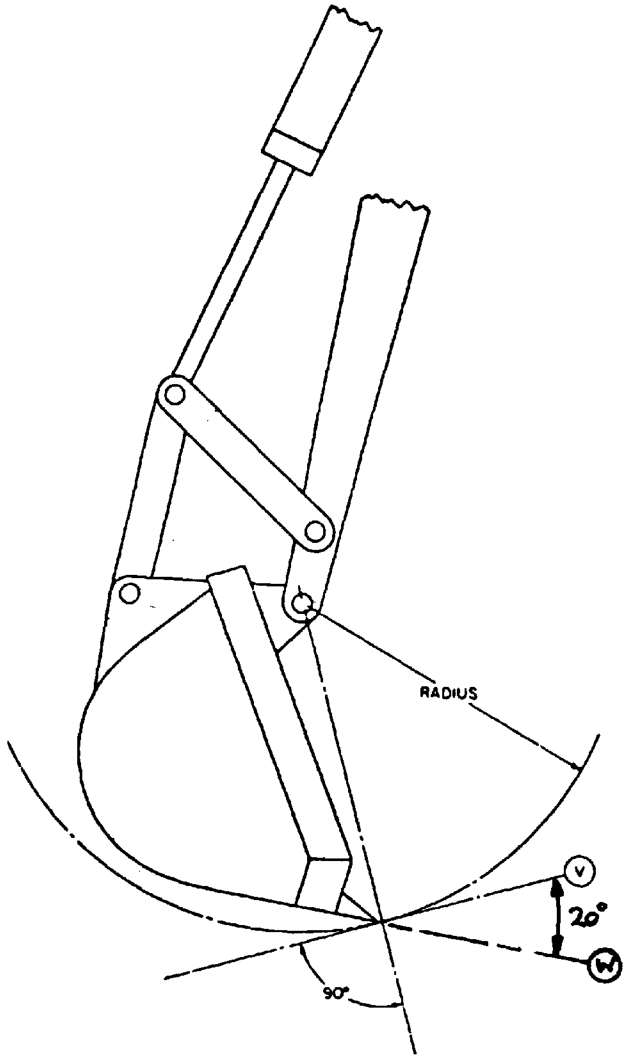


FIGURE 3.8

Definition of Maximum Bucket Tooth Force -

ICED Standard (ICED[11])

Thus the maximum bucket tooth force will depend upon which hydraulic cylinder reaches its maximum force first, for a given geometric configuration.

3.2.3 Service Line Relief Valve Efficiency

Throughout the complete range of geometric configurations, each of the service line pressure relief valves come into operation limiting the maximum bucket tooth force. In order to assess the efficiency of each of the service line pressure relief valves, the number of times each comes into operation for a complete range of configurations can easily be determined as follows:-

$$f_{RAM1} = \frac{\text{No. of Configs.in which bucket tooth force is limited by Ram1 SLRV}}{\text{Total no. of geometric configurations}} \tag{3.78}$$

Similarly criteria may be applied to f_{ram2}, f_{ram3} etc such that:-

$$f_{RAM1} + f_{RAM2} + f_{RAM3} = 1 \tag{3.79}$$

These may be also expressed as a percentage.

3.2.4 Digging Performance

The digging performance of an excavator is best described in terms of the ICED bucket tooth force rating(ICED[13]). Because of the nature of the linkage mechanism it is possible during a range of geometric configurations to obtain a high maximum bucket tooth force but low average bucket tooth force. It is therefore important to quantify both the maximum and average bucket tooth forces for a particular service line relief valve combination and range of geometric configurations these are defined as follows.

Fmax = maximum recorded bucket tooth force achievable over the entire range of geometric configurations.

$$Favge = \frac{\sum \text{Max tooth force for each geometric configuration}}{\text{Total No. of geometric configurations}} \tag{3.80}$$

3.2.5 Working Safety Factor

The structural strength of the excavator digging arm is best described by the 'working' safety factor for each location of interest on the structure. The safety factor S.F. is given by:-

$$S.F. = \frac{\sigma_y}{\sigma_c} \qquad (3.81)$$

Where σ_y is the material yield stress
and σ_c is the combined stress at the location of interest

Throughout the range of geometric configurations during a typical digging cycle, the forces at the joints and consequently the combined stresses at particular locations will be continually changing. Hence the safety factor S.F. as defined will also be changing. To determine the overall 'working' safety factor, the combined stress at all locations of interest must be found for the entire range of geometric configurations. Having determined the safety factors for all possible geometric configurations, the true 'working' safety factor is the minimum recorded safety factor.

3.2.6 Design Optimisation Parameters

The major factor considered when formulating the design specification of a microexcavator is its performance. A number of inter-related design parameters must be determined in order to achieve an acceptable digging/lift performance with a sound structure. There are generally two groups of design parameters for consideration, those relating to the mechanism and structure and those related to the hydraulic system (Figure 3.9). The 'matching' of a hydraulic system and structure is extremely important. Incorrect design of the hydraulic system could overstress certain regions of the digging arm mechanism or under utilise the structural strength when better digging performance could be achieved. For the application of this design optimisation approach to the existing Powerfab 360 digging arm structure, a number of optimisation parameters have already been 'fixed' and are not easily changed such as structural design, linkage geometry and hydraulic cylinder dimensions (Figure 3.9). A hierarchial structure can be drawn to illustrate the cost and complexity of changing certain design parameters as shown overleaf.

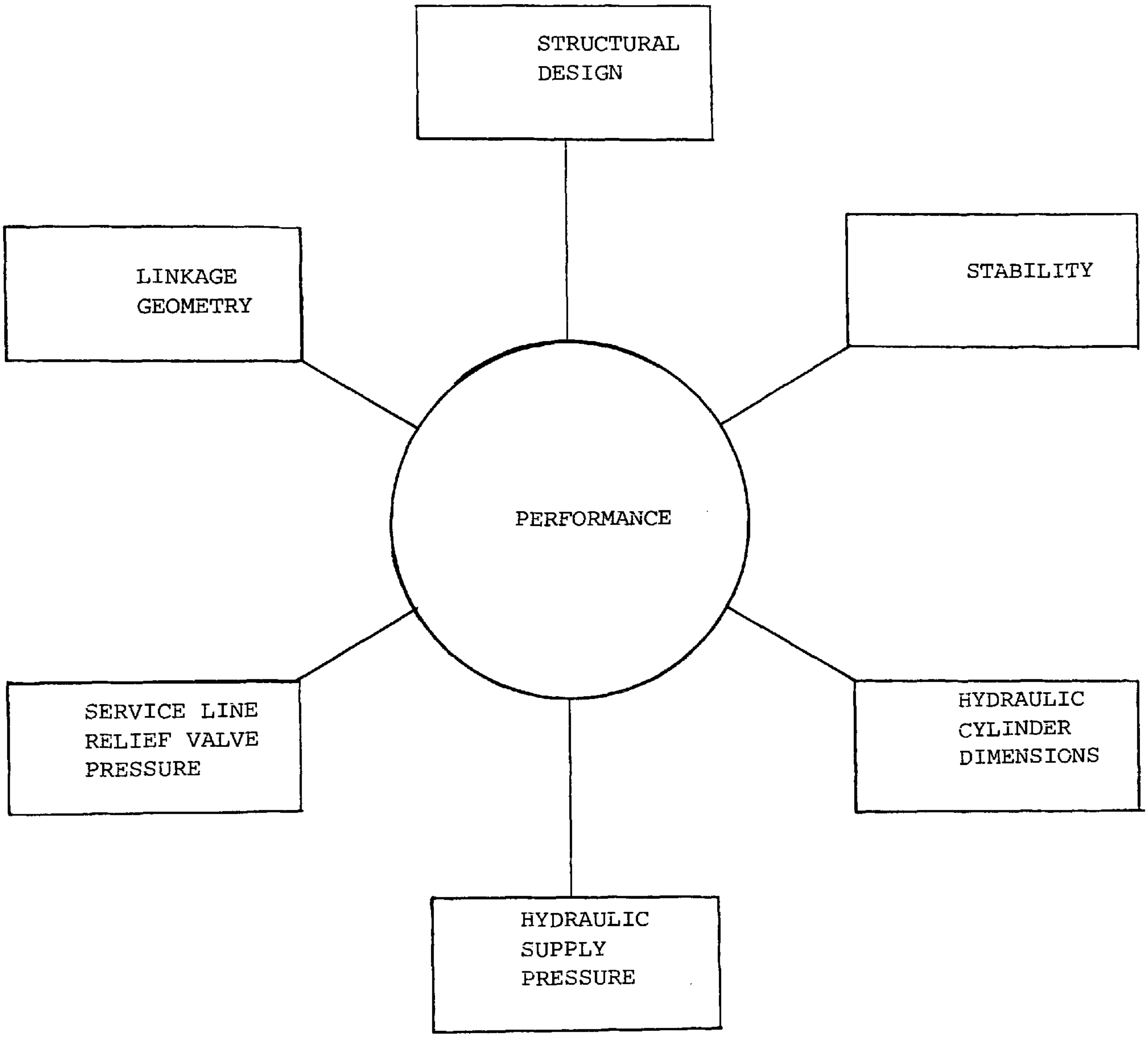
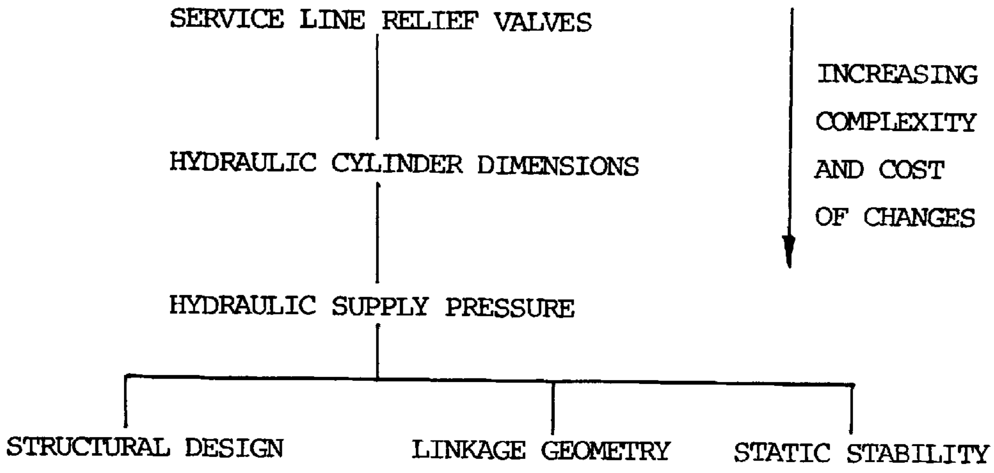


FIGURE 3.9

Design Optimisation Parameters



Thus for an existing structure the starting point for design optimisation is service line relief valve pressure settings, followed by hydraulic cylinder sizes and combination etc.

3.2.7 Setting the Optimisation Parameters

In order to carry out a design optimisation exercise two factors must first be established, these are an acceptable digging performance and secondly an acceptable working safety factor. As previously discussed (3.2.4), the digging performance is best described in terms of the average and maximum bucket tooth force. Presenting the data in graphical format gives an instant indication if the design factors are satisfied or not. Graphs of working safety factor against average digging force and against maximum digging force are necessary as shown in Figures 3.10(a), (b).

An 'operating point' on the graph represents the digging force and overall minimum safety factor for a particular combination of service line relief valve settings. An acceptable safety factor is represented by a horizontal line on the graph (Figures 3.10(a), (b)). Any operating points that lie above this line represent an acceptable working safety factor and any points below are unacceptable; similarly a vertical line is used to represent an acceptable average or maximum digging force (Figure3.10(a),(b)).

The point of intersection p, represents the 'optimum' operating point.

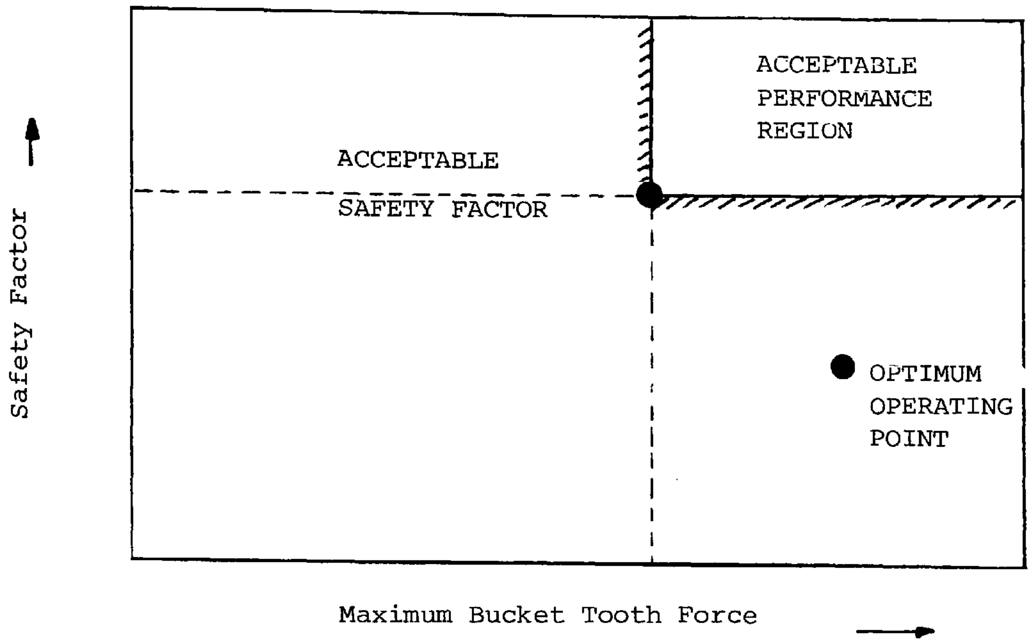


FIGURE 3.10(a)

Variation of Safety Factor with Maximum Bucket Tooth Force

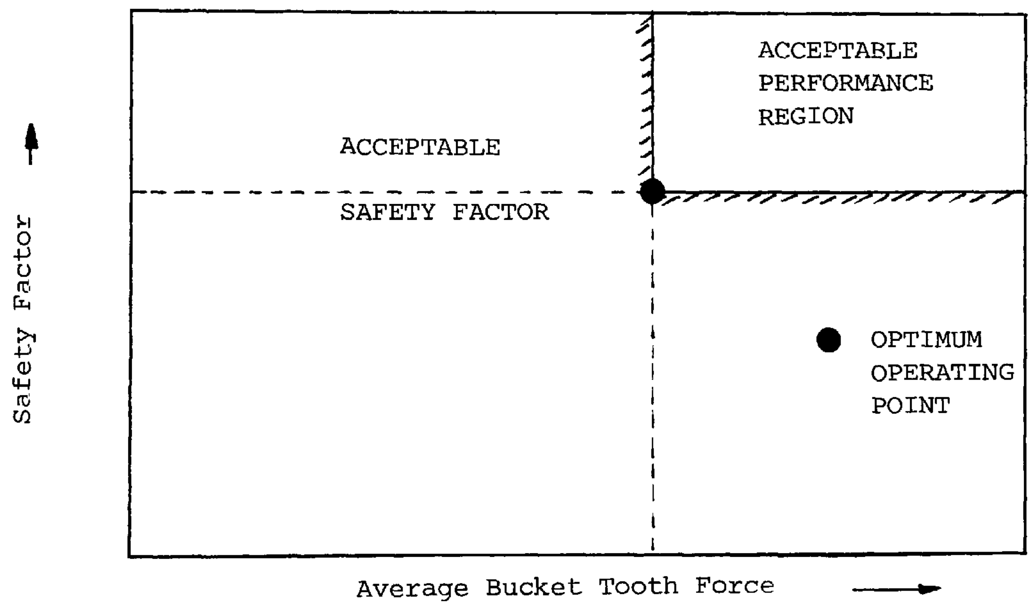


FIGURE 3.10(b)

Variation of Safety Factor with Average Bucket Tooth Force

FIGURE 3.10

Design Optimisation Plots

3.3 COMPUTER AIDED DESIGN SOFTWARE

The computer theory developed in sections 3.1 and 3.2 was used to develop an integrated CAD software package to allow the design engineer to carry out structural analysis and design optimisation calculations on the Powerfab 360WT microexcavator.

3.3.1 Software Documentation and Design

At the outset of the project detailed software documentation standards were written. Rigid standards are necessary for software projects that require future software support and involve more than one programmer analyst. Common standards ensure that high quality structured software is developed. Appendix 1 shows the software documentation standards that were written using references Koptez[15], Sommerville[16], Zaresiti[17], Fox[18], Zeikowitz[19].

The software documentation standards require detailed user guides and program specifications to be written before any program coding is commenced. Appendix 2 shows the user guide and program specification developed for Program P360, one of the CAD software modules.

3.3.2 CAD Programs

The CAD software package comprises of seven separate programs (Figure 3.11) because of the memory and speed restrictions of the BBC 128K master microcomputer, the programs were written in blocks of 32K. Inter-communication of the programs was achieved by using common data files held on the program disk. The programs were written using structured BBC Basic, Coll[14]. Each program is described in detail in the following sections.

GEN01
This program can be used to calculate the section properties of complicated section shapes. The traditional hand calculation approach is often time consuming and complex. This program uses a swept vector approach Podmore[20], the section surfaces are represented by a series of straight lines.

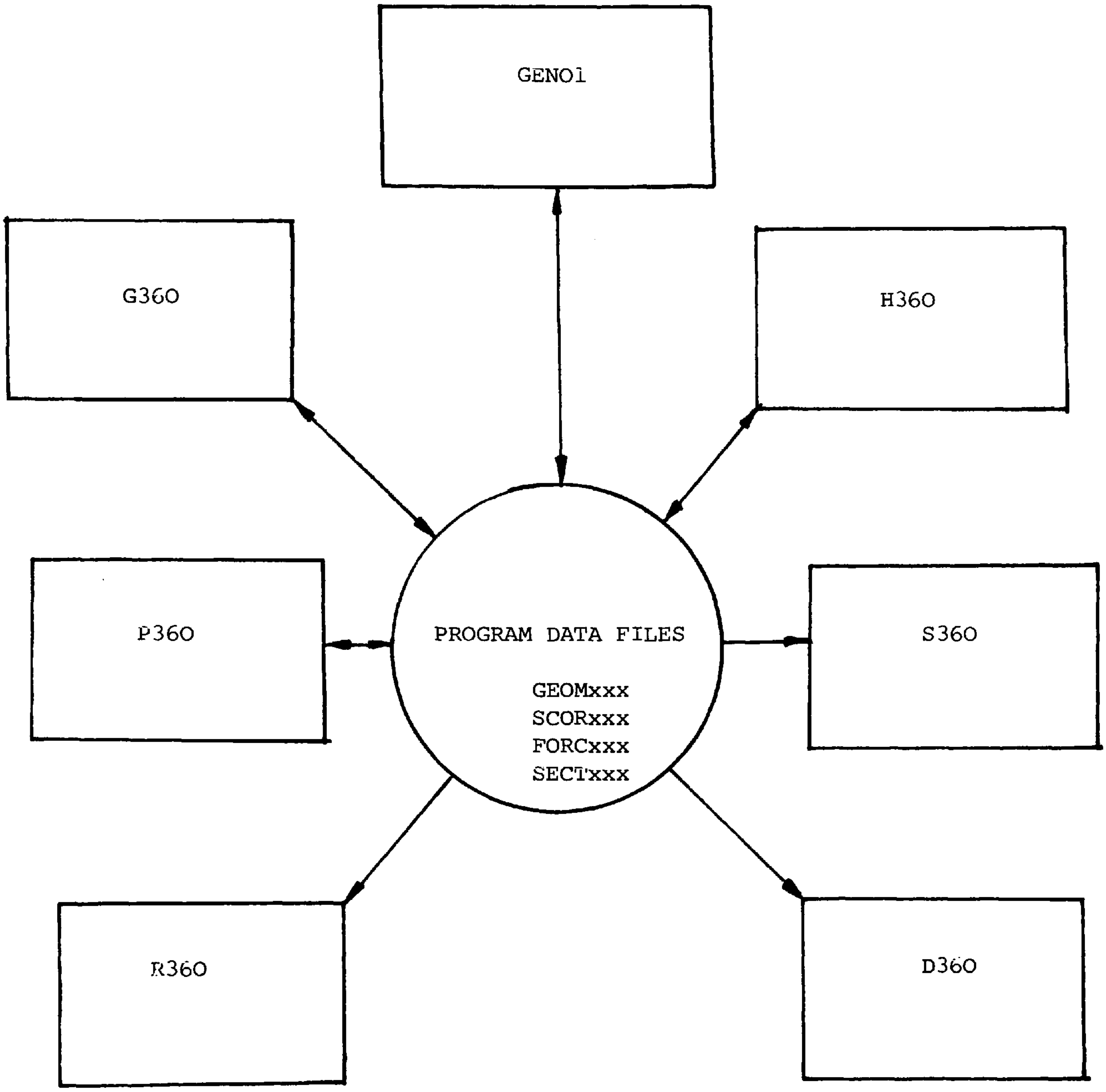


FIGURE 3.11

Intercommunication of CAD Software

The program can be used to create new sections, save section to disk or load a section from disk. A hard copy of he section and section properties may also be obtained if necessary.

G360

This program is used to create and modify a geometry data file for use by other CAD programs in the Powerfab 360 project. The geometry data file contains all the rigid link dimensions for the Powerfab 360 digging arm. These link sizes are fixed for a particular machine but may be changed at any time by the user as necessary.

The program allows the user to create a new geometry data file from scratch, load an existing file from disk save a file to disk or print out the data file for a permanent record.

H360

This program is used to create and modify a section properties data file for use by other programs in the Powerfab 360 project. Having selected the points of interest for stress analysis on the Powerfab 360 digging arm, the following data is required:-

1. Distance of point of interest from a convenient datum (x).

2. Second moment of area of the section (Ixx).

3. Distance of outermost fibre from the neutral axis (y).

4. Cross-sectional area A(xx).

Up to 20 locations may be selected on the dipper arm, upper boom and lower boom.

The user may create a new section properties data file, save a file to disk, load a file from disk or print out the data file for permanent hard copy.

P360

This program allows the user to calculate the magnitude and direction of all the forces on the linkages of the Powerfab 360 digging arm. It uses a data file containing the major dimensions of the linkages (GEOMXXX) to carry out the calculations and outputs the force data to another data file (FORCXXX).

The user may select which units he wishes to work in (metric/imperial) and may calculate forces for a single of range of geometric configurations defined by the open, closed lengths and extension increments of each of the three hydraulic cylinders.

In addition to entering this data the user is also required to specify the magnitude and direction of the bucket tooth force. When the program is required to create a data file for the calculation of maximum tooth forces or stresses (using programs R360, D360) the user must input a unit tooth force.

S360

This program allows the user to calculate stresses at various specified points of interest on the digging arm. Direct bending and combined stresses are found for all locations. The basic dimensions of the linkages are obtained from data file GEOMXXX. The points of interest are obtained in a data file SECTXXX and the forces on the structure in data file FORCXXX, these files must be created before the program can be used. For a given material type the safety factor for each location is calculated and displayed. The calculations are performed for a single or range of geometric configurations.

R360

This program allows the user to determine the digging performance of a microexcavator. The program requires the service line relief valve (SLRV) pressures and hydraulic cylinder dimensions to be set up initially. The program requires a data file of forces for a range of geometric configurations specified by forces data file FORCXXX. This file must be created by program P360 using a UNIT tooth force (ICED standard).

For each geometric configuration the program calculates the achievable bucket tooth force using all three hydraulic cylinders. This bucket tooth force is limited by the SLRV pressure setting. The SLRV efficiency, maximum and average digging forces are also calculated by the program. All data is displayed on the screen but may also be printed if required.

D360

This program can be used to calculate the theoretical maximum digging forces, associated stresses and safety factors at pre-determined positions for the three arms of the Powerfab 360 microexcavator. These calculations are possible for:-

(a) a range of pre-determined arm positions
(b) a variable combination of hydraulic cylinder dimensions
(c) a variable set of materials and associated yield stresses
(d) a range of hydraulic relief valve settings, or fixed values as required.

During run time the program extracts information from three data files which contain the information regarding sectional properties, linkage dimensions and force data for the points of interest. These files are created external to the main program.

A hard copy of the results can be produced which gives the following information:-

(a) relief valve settings
(b) data for a complete sweep of the geometry
(c) safety factor data

It should be noted that run time can be extremely long. The run time is dependent upon the number of points of interest, number of ram positions considered and the range of relief valve setting chosen.

The program is designed to be self-explanatory, being totally menu-driven.

CHAPTER FOUR

4. EXPERIMENTATION

Experimental measurements are necessary to evaluate the accuracy of the computer modelling methods; the experimental data can provide useful results for computer model development and give practical results in addition to the theoretical results of the model.

An experimental method was developed to allow stresses at locations of interest and bucket tooth forces to be measured in a range of geometric configurations. The number of geometric configurations were limited only by the physical restrictions of the loading frame used.

4.1 Experimental Apparatus

The experimental apparatus was designed to allow measurements of the following parameters:-

1. Magnitude and direction of bucket tooth forces.

2. Magnitude and direction of stress at locations of interest on the structure.

3. Hydraulic cylinder lengths for pre-determined geometric configuration.

The Powerfab 360WT Microexcavator was fixed in a large rigid structural loading frame to allow measurements to be taken (Plate 4.1). The front outrigger stabiliser extensions were removed and the front outriggers pinned to the centre section of the loading frame. The rear outrigger stabiliser extensions were fitted with rubber feet and allowed to rest on the laboratory floor. The lower platform was also bolted to the loading frame hence the microexcavator was rigidly fixed.

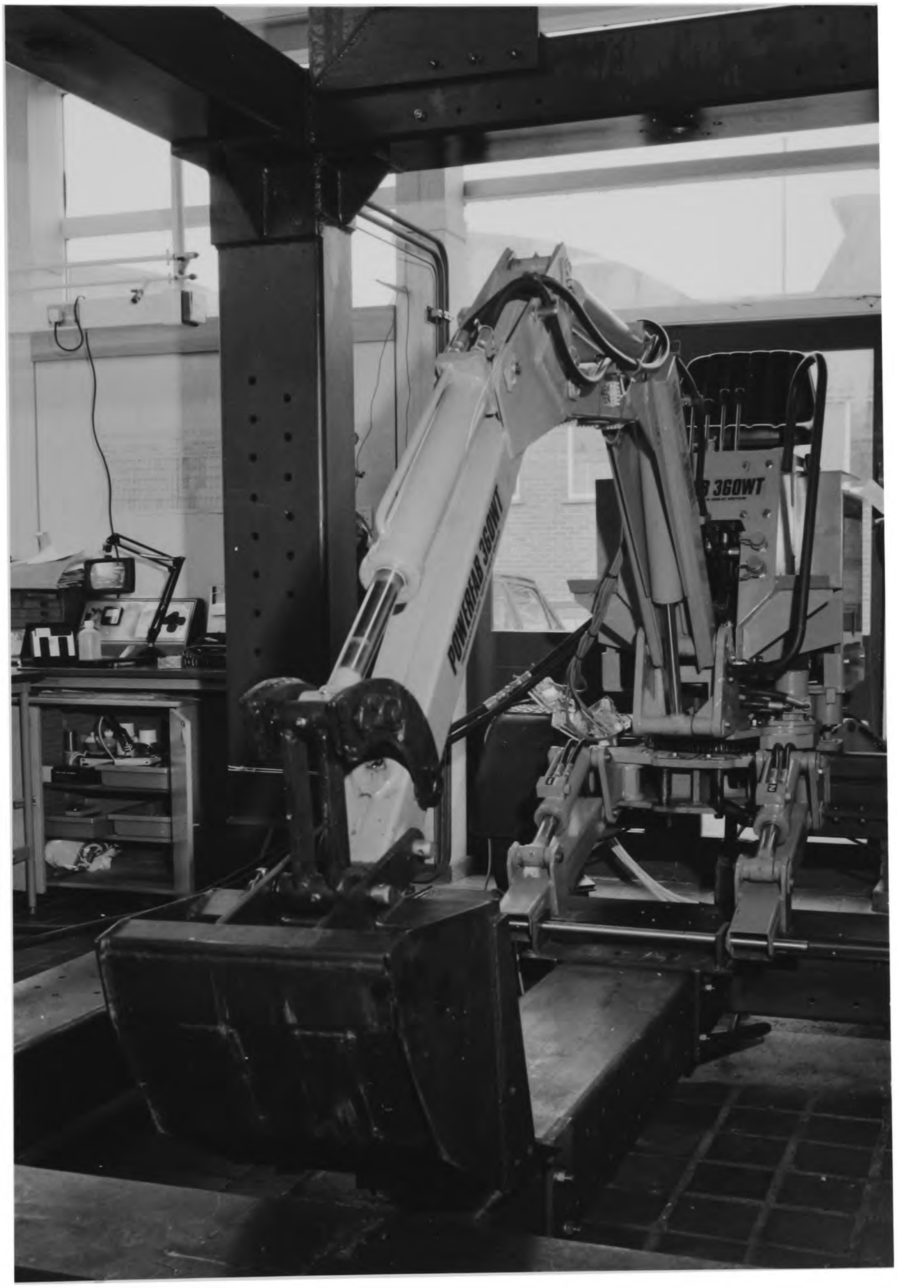


PLATE 4.1
Powerfab 360WT Microexcavator Mounted in the Loading Frame

4.1.1 Measurement of Bucket Tooth Forces

Specially designed bucket tooth and load cells (Plate 4.2) were used by O'Brien[3] and Stephens[4] to measure the actual bucket tooth forces applying a load using one or more hydraulic cylinders whilst the microexcavator was fixed in the loading frame.

The load cells were first calibrated using a large loading machine and a calibration factor obtained, Thomas[20].

One load cell was used to measure the load in the X-direction parallel to the horizontal and the other to measure the load in the Y-direction parallel to the vertical (Plate 4.3).

The magnitude and direction of the resultant bucket tooth force is then given by:-

$$\text{Fres} = \sqrt{F_x^2 + F_y^2} \tag{4.1}$$

$$\text{angres} = \text{TAN}^{-1} \left(\frac{F_y}{F_x}\right) \tag{4.2}$$

Figure 4.1 shows the load cell arrangement for measuring bucket tooth force.

A number of inaccuracies were identified with this method of measuring bucket tooth force. These include restriction of rotation of the bucket tooth which influenced the accuracy of the x-y loads and hence the direction of the resultant tooth force (Plate 4.3).

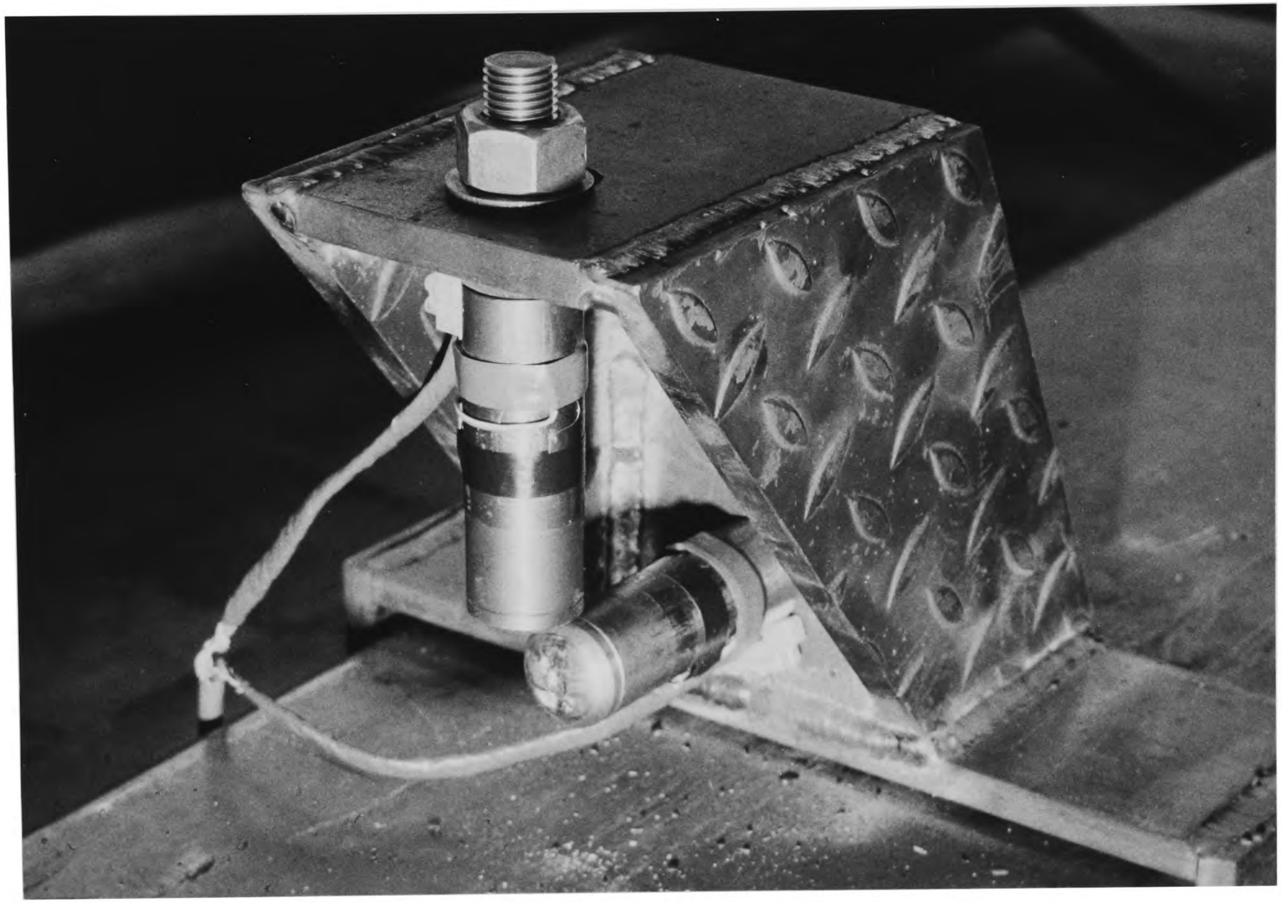


PLATE 4.2

Load Cells Mounted on the Loading Frame

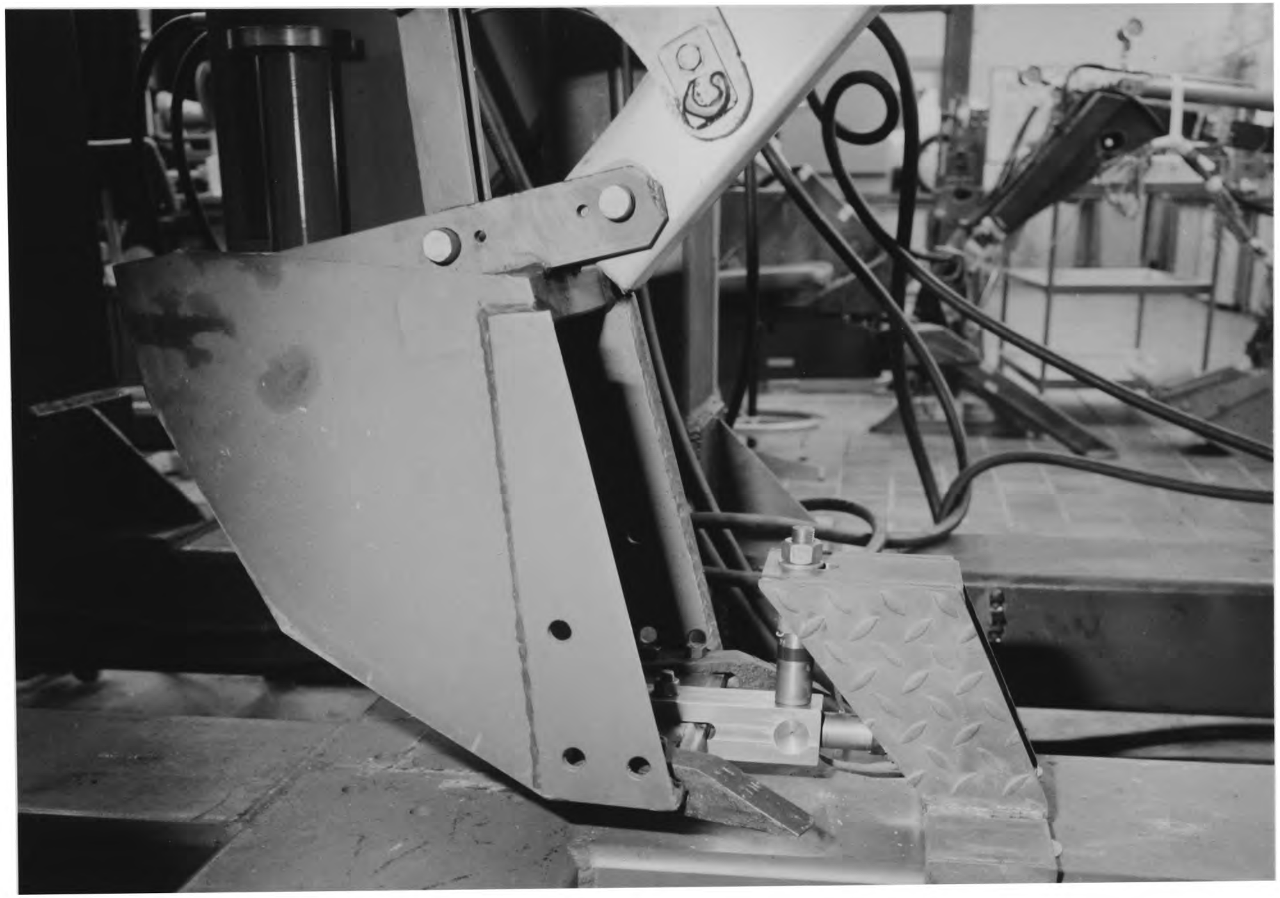


PLATE 4.3

Original Bucket Tooth Mounted on the Load Cells

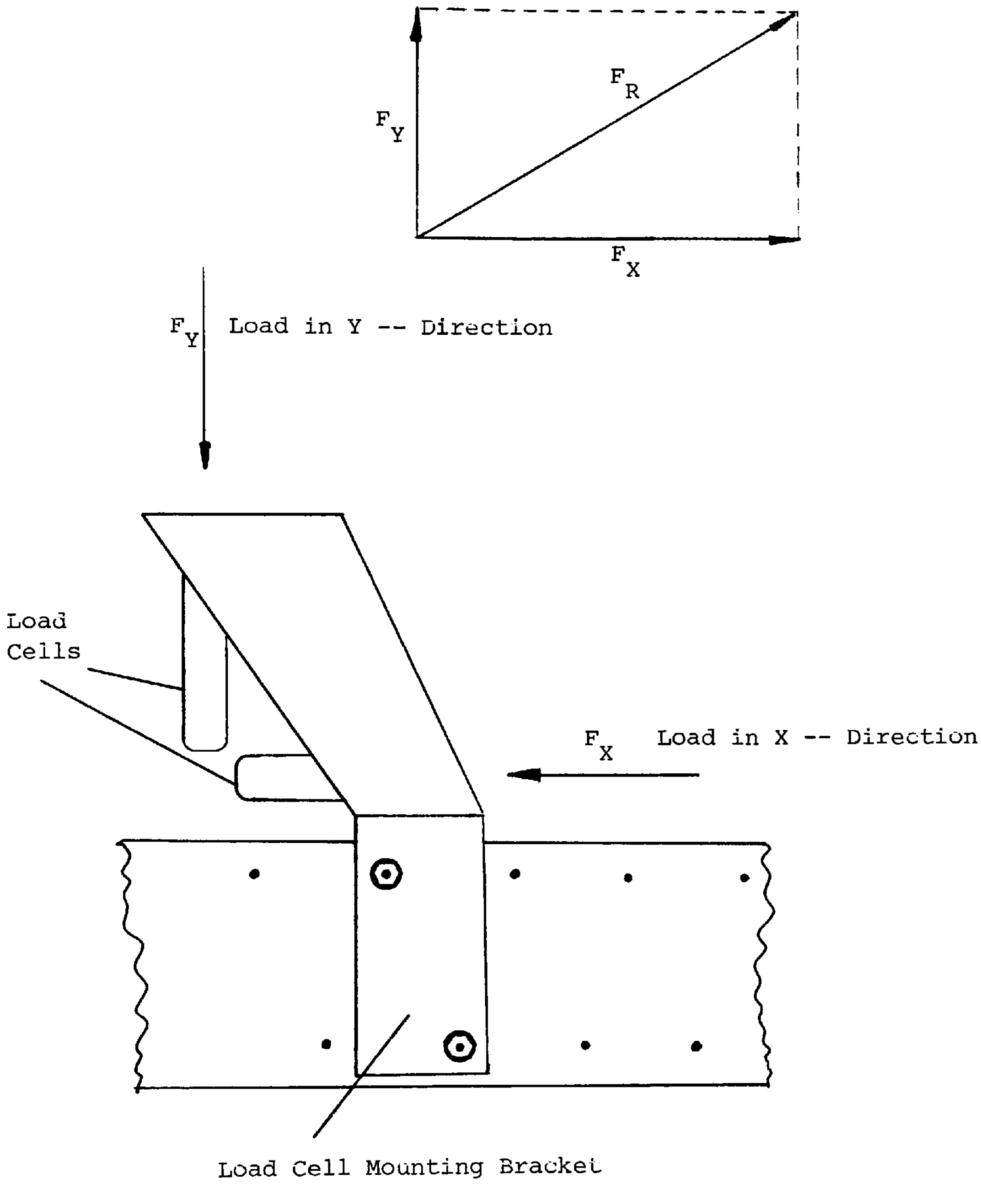


FIGURE 4.1

Measurement of Bucket Tooth Force

4.1.2 Revised Measurement of Bucket Tooth Forces

The recessed bucket tooth was replaced by a 50mm diameter roller and pin free rotation through 360° was thus possible (Plate 4.4). The bearing surface was well greased to keep friction at the roller to a minimum. The virtual point of measurement of the bucket tooth force is thus the actual centre line of the pivot pin.

4.1.3 Measurement of Stress

Previous practical measurements of stress on the Powerfab 125 Microexcavator (O'Briens[3], Stephens[4]), were made using a combination of linear and rosette strain gauges. The conclusion drawn from these results was that rosette gauges gave more accurate stress results providing both the magnitude and direction of stress at the location of the strain gauge. For the series of tests on the Powerfab 360WT Microexcavator three element -45°, 0, +45° strain gauge rosettes were used (Plate 4.5). The strain gauge rosettes measure only strain values in micro-strain. These must be processed to obtain actual stress direction and magnitude (shear stress).

4.1.4 Data Acquisition and Analysis System

Previous experimental data acquisition was achieved using an Orion Delta datalogger unit only. The measured strain values were converted to actual stresses and loads using the basic programming facility of the datalogger.

This system proved to be slow and the basic programs difficult to modify since the basic language used by the datalogger is primitive. To overcome these problems a rapid data acquisition system was developed with the assistance of an engineering student Mr. D.P. Thomas[21]. The Orion Delta datalogger was used for rapid data acquisition and the BBC microcomputer for the analysis of results, Thomas, Bromfield and Evans[22].

The load cells and strain gauges were connected to the datalogger to complete the system (Plate 4.6).

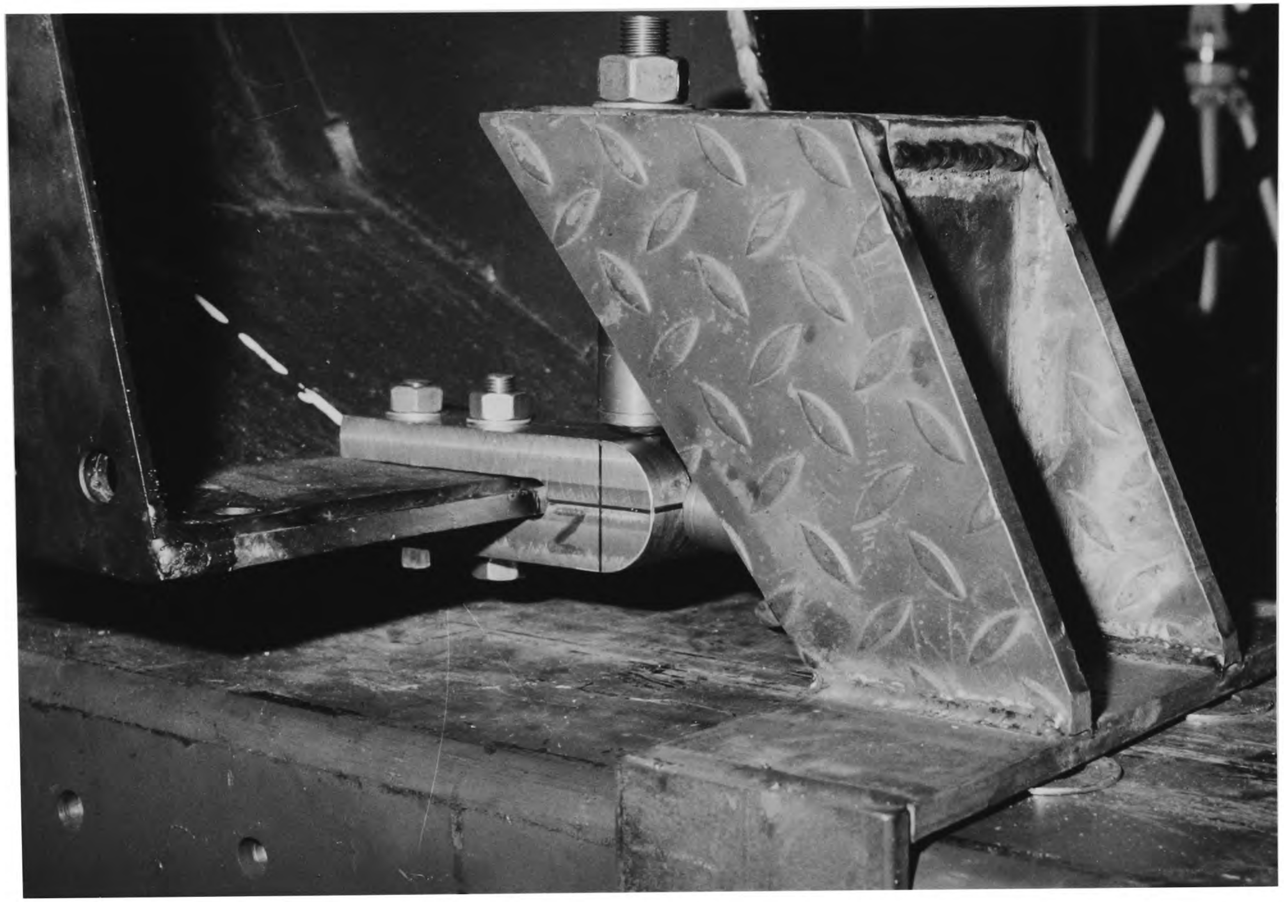


PLATE 4.4

Modified Bucket Tooth Mounted on the Load Cells

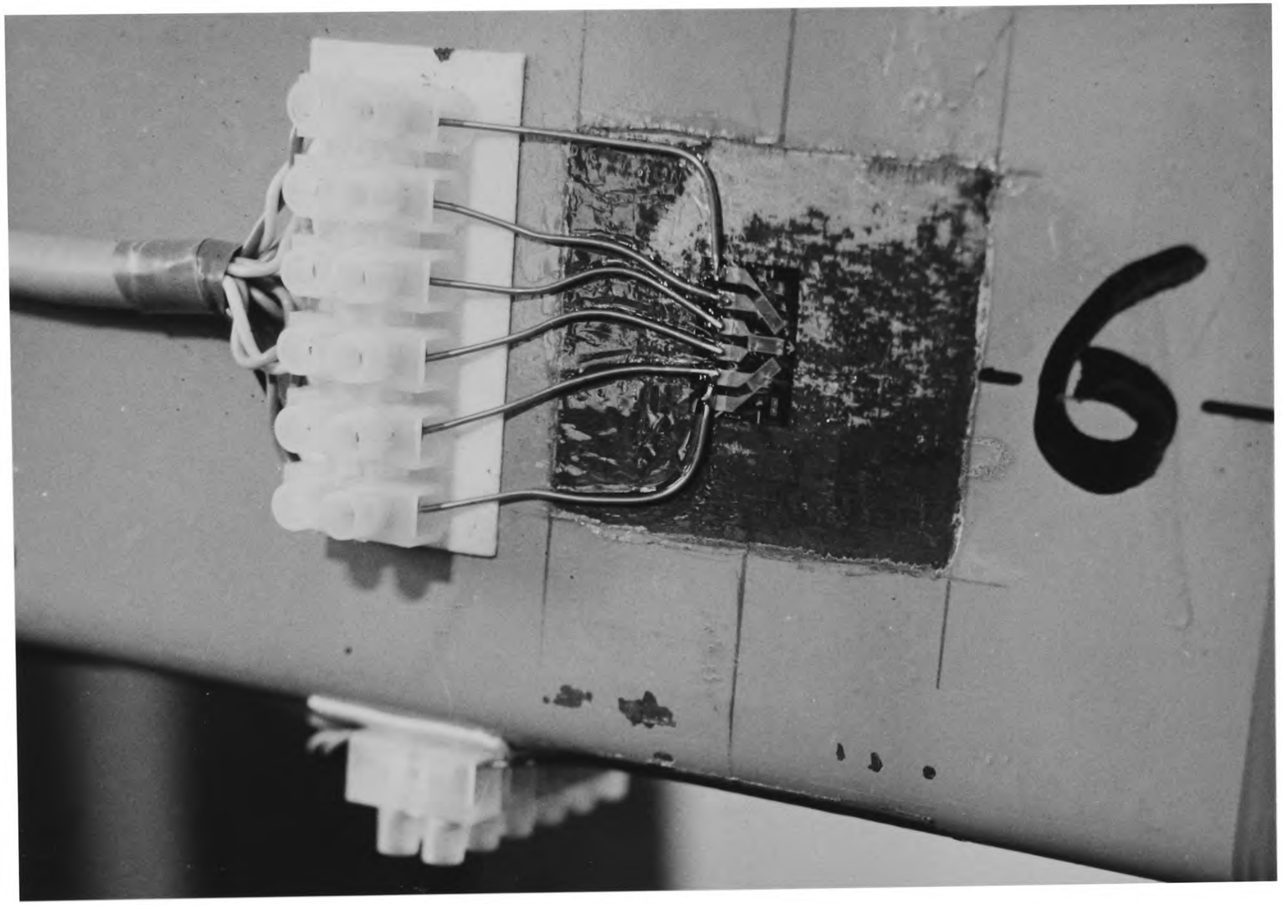


PLATE 4.5

Three Element Electrical Resistance Strain Gauge Rosette

PLATE 4.6

Data Analysis and Acquisition System

4.2 Experimental Method

It was decided initially to investigate the stresses on the upper boom section of the digging arm only, due to time and cost restraints of complete structural testing of all of the major components of the digging arm.

4.2.1 Strain Gauge Application

Electrical resistance three element strain gauge rosettes were bonded to upper boom section at 8 locations in total on the upper surface, sides and lower surface of the upper boom (Figure 4.2). The strain gauges on the upper and lower surface were mounted off set from the centre line of the section due to the fillet weld joining the two halves of the upper boom (Plates 4.7 and 4.8).

4.2.2 Hydraulic Supply

In order to avoid using the Microexcavator petrol engine and hydraulic power unit, the control valve bank inlet and outlet were connected to the laboratory hydraulic power supply, a pressure gauge and flow control valve were connected in line to regulate and monitor the supply.

The hydraulic supply pressure was set at 176 bar (2600 psi) and the flow rate 21.5 litres/minute (4.7 gallons/min); these figures represent the maximum values encountered during normal operating conditions.

4.2.3 Configuration for Stress and Bucket Tooth Force Measurement

Having fixed the Microexcavator lower platform in the loading frame the geometric configuration for measurements was fixed by the location of the load cell mounting bracket. The bracket straddles the central beam of the loading frame and is fixed in place by two bolts through the section. Three discrete configurations were chosen for measurements as shown in Figure 4.3.

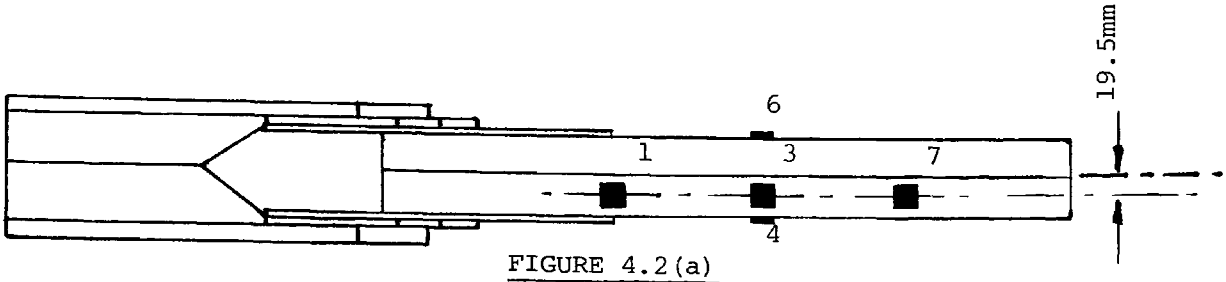


FIGURE 4.2(a)

View from Above

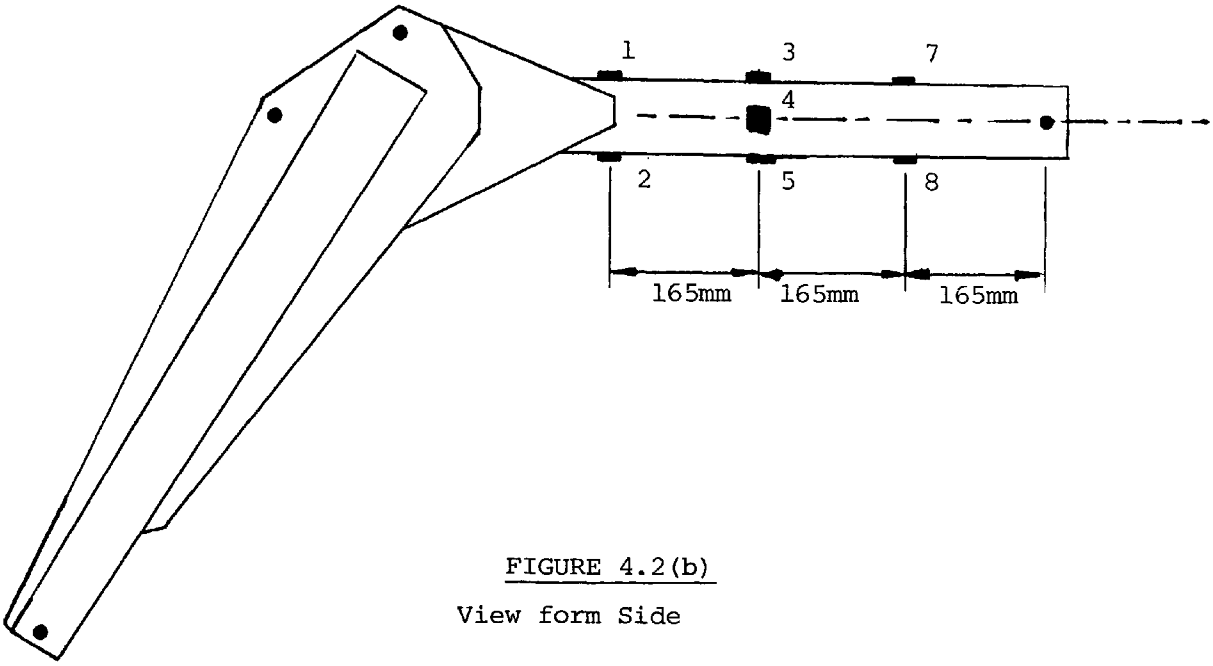


FIGURE 4.2(b)

View form Side

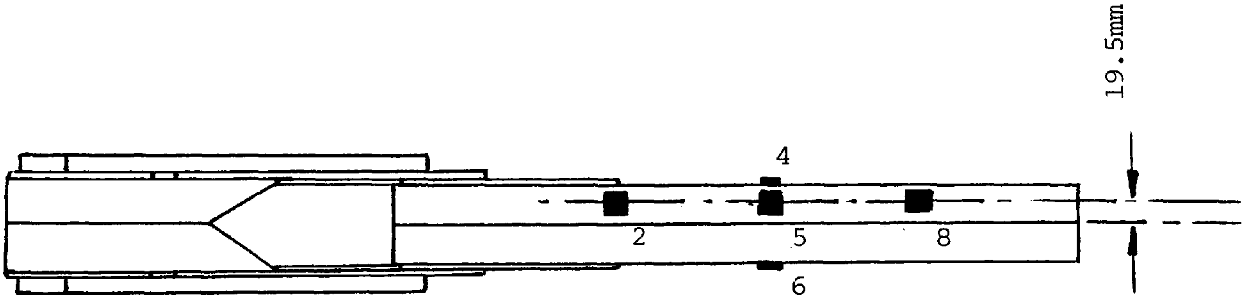


FIGURE 4.2(c)

View form Below

FIGURE 4.2

Location of the Strain Gauges on the Boom Assembly

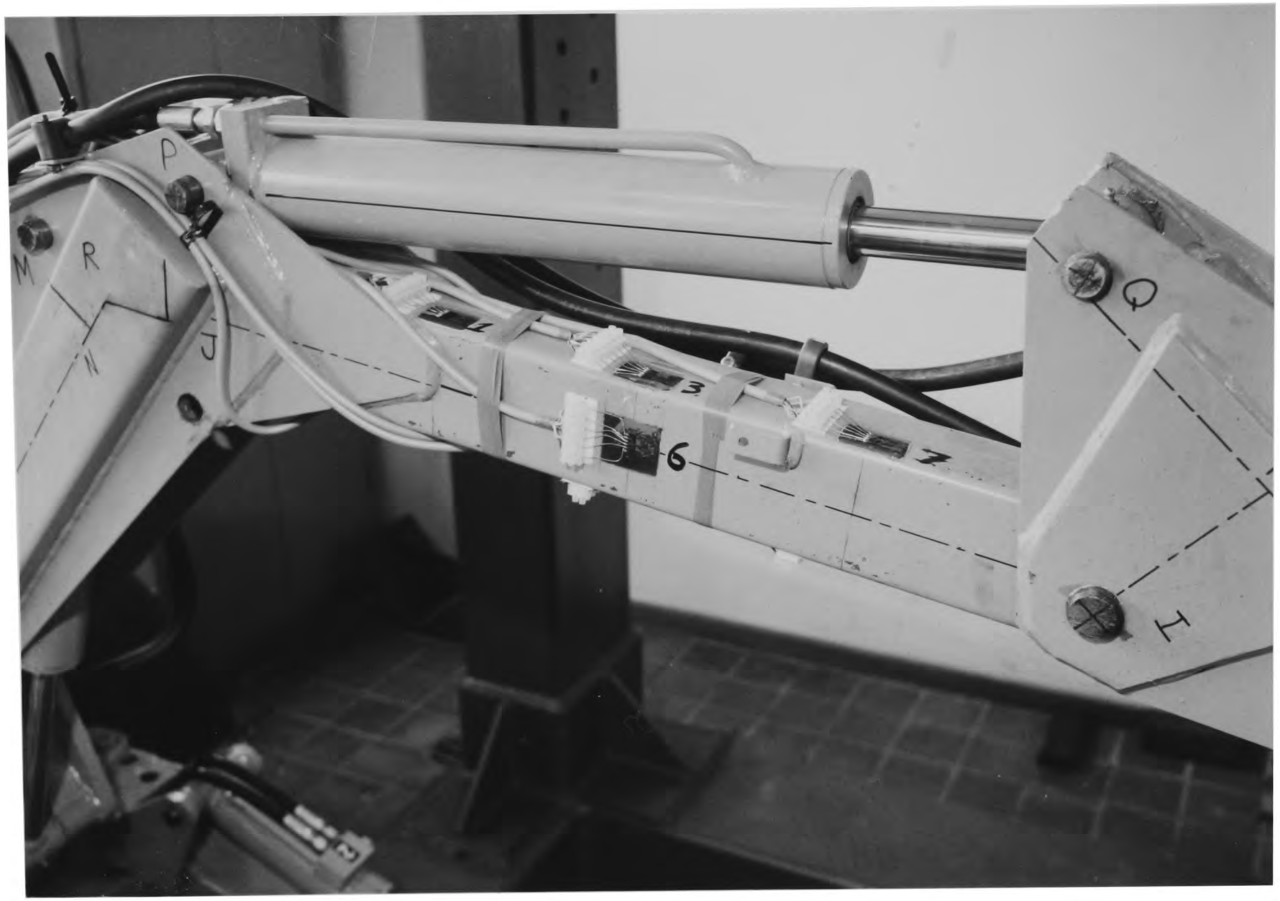


PLATE 4.7

Location of the Strain Gauges on the Upper Boom (Upper Surface)

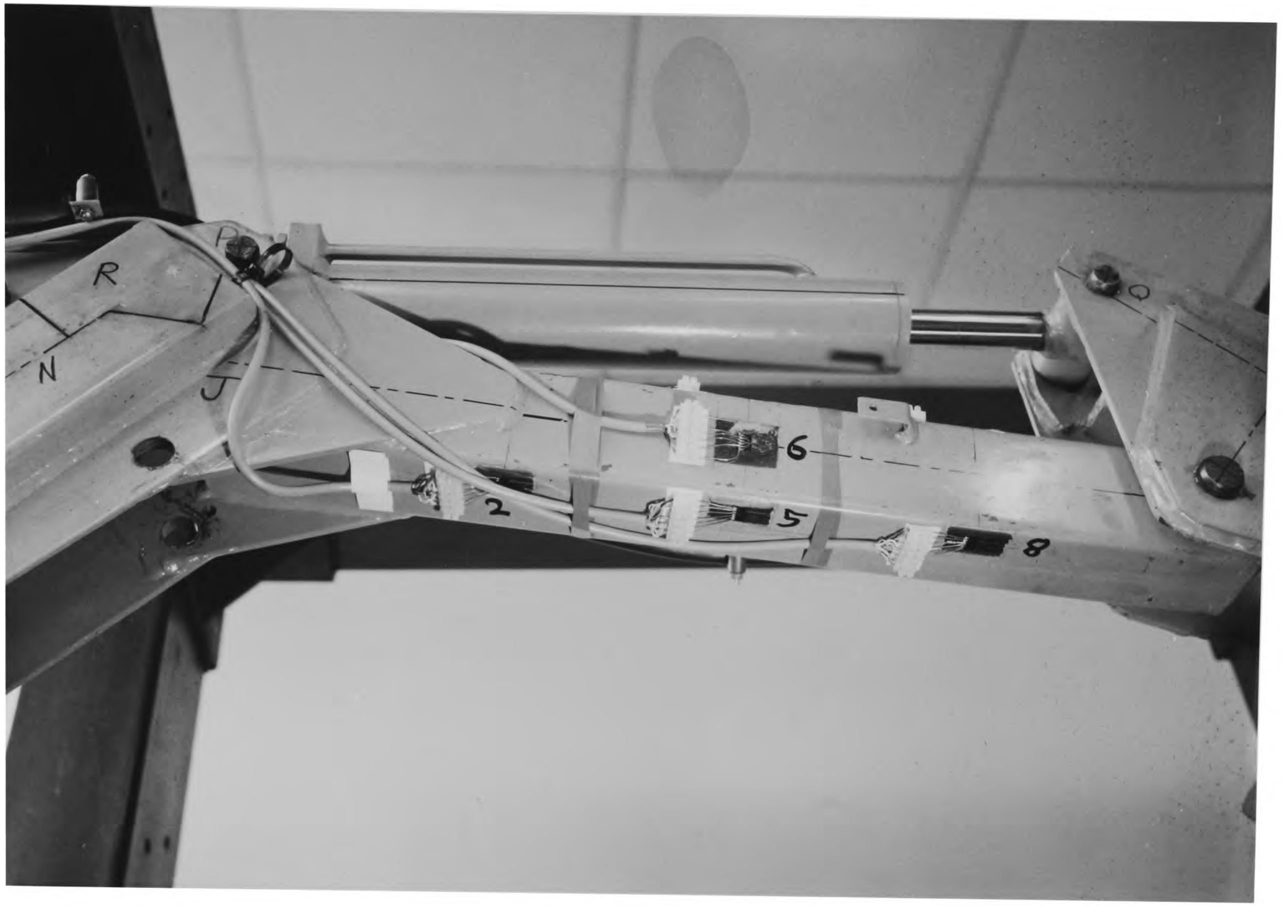


PLATE 4.8

Location of the Strain Gauges on the Upper Boom (Lower Surface)

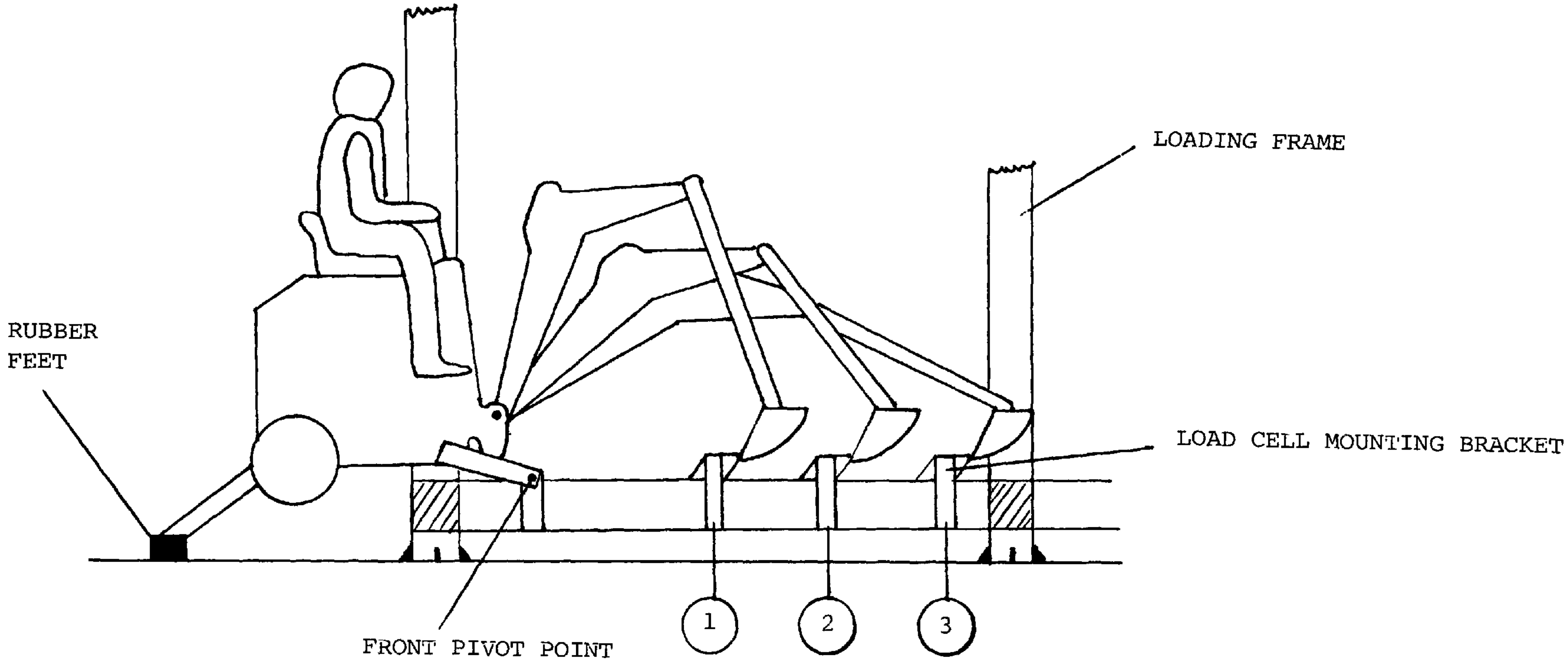


FIGURE 4.3

Geometric Configurations for Experimental Measurements

Once the load cell bracket had been fixed in position the Microexcavator digging arm was manoeuvred to locate the bucket tooth close to the load cells and as near to the horizontal as possible. At this point the data acquisition system is run to obtain the 'Zero' strain readings for the datalogger. The structural weight of the digging arm is then ignored during subsequent measurements. Having initialised the strain gauges a load is applied at the bucket tooth onto the load cell by the operator using either the bucket or dipper hydraulic cylinder via the operating lever. Only one hydraulic cylinder may be used at a time since the movements of the bucket tooth is difficult to predict and control using more than one hydraulic cylinder. The boom cylinder could not be operated to induce a load because it was near its fully open state while in the loading frame.

4.2.4 Measured Parameters

In addition to the measurements of stress and bucket tooth force using the data acquisition and analysis system, other parameters that were measured were:-

1. Hydraulic cylinder lengths (mm) - using a meter ruler.

2. Horizontal attitude of the bucket tooth - using a spirit level and engineering protractor.

The measurement of the hydraulic cylinder lengths is necessary to define the geometric configuration for measurement. Each unique configuration will have its own unique combination of three hydraulic cylinder lengths. These dimensions are used as input data for the computer model described in Chapter 3.

The measurement of the horizontal attitude of the bucket tooth is necessary to determine the angle of inclination of the bucket tooth force as measured by the load cells, to the bucket tooth itself. The magnitude of the bucket tooth force and the angle of inclination to the bucket tooth are also used as input parameters to the computer model.

CHAPTER FIVE

5. THEORETICAL AND EXPERIMENTAL RESULTS

This chapter presents and discusses the experimental and theoretical structural analysis results and also the theoretical design optimisation results.

For the structural analysis of the microexcavator, theoretical and experimental stress results are presented for one typical configuration using both the bucket and dipper cylinders independently to load the digging arm. These stress results are shown for the original bucket tooth arrangement, the modified bucket tooth arrangement, and for the modified bucket tooth arrangement with a correction factor for the effects of torsion. The initial assumptions that were made about the structural model are reviewed.

The computer method used to obtain the design optimisation results is briefly described and these results are presented in two forms and discussed.

5.1 Structural Analysis Results

Three geometric configurations were used for the experimental measurements, as detailed in chapter 4 and shown in Figure 4.3.

For each geometric configuration bucket tooth forces were applied to the load cells using either the bucket or dipper hydraulic cylinder independently. For each configuration and hydraulic cylinder used, a graph of the experimental combined stress distribution for the upper, lower and side surfaces was plotted.

The measured hydraulic cylinder dimensions and the magnitude and direction of the bucket tooth force were recorded for input to the computer model. Program G360 was then used to set up the dimensions of the rigid links representing the digging arm structure. Program H360 was used to specify the section properties for the locations of interest for stress calculations.

The recorded hydraulic cylinder lengths and the magnitude and direction of the bucket tooth force were then inputted to Program P360 and the magnitude and direction of the forces at the joints calculated. This data was then used by Program S360 to calculate the combined stress distribution on the upper boom section. This stress distribution was superimposed on the experimental stress distribution plot for comparison.

5.1.1 Theoretical and Experimental Results - Original Bucket Tooth Arrangement

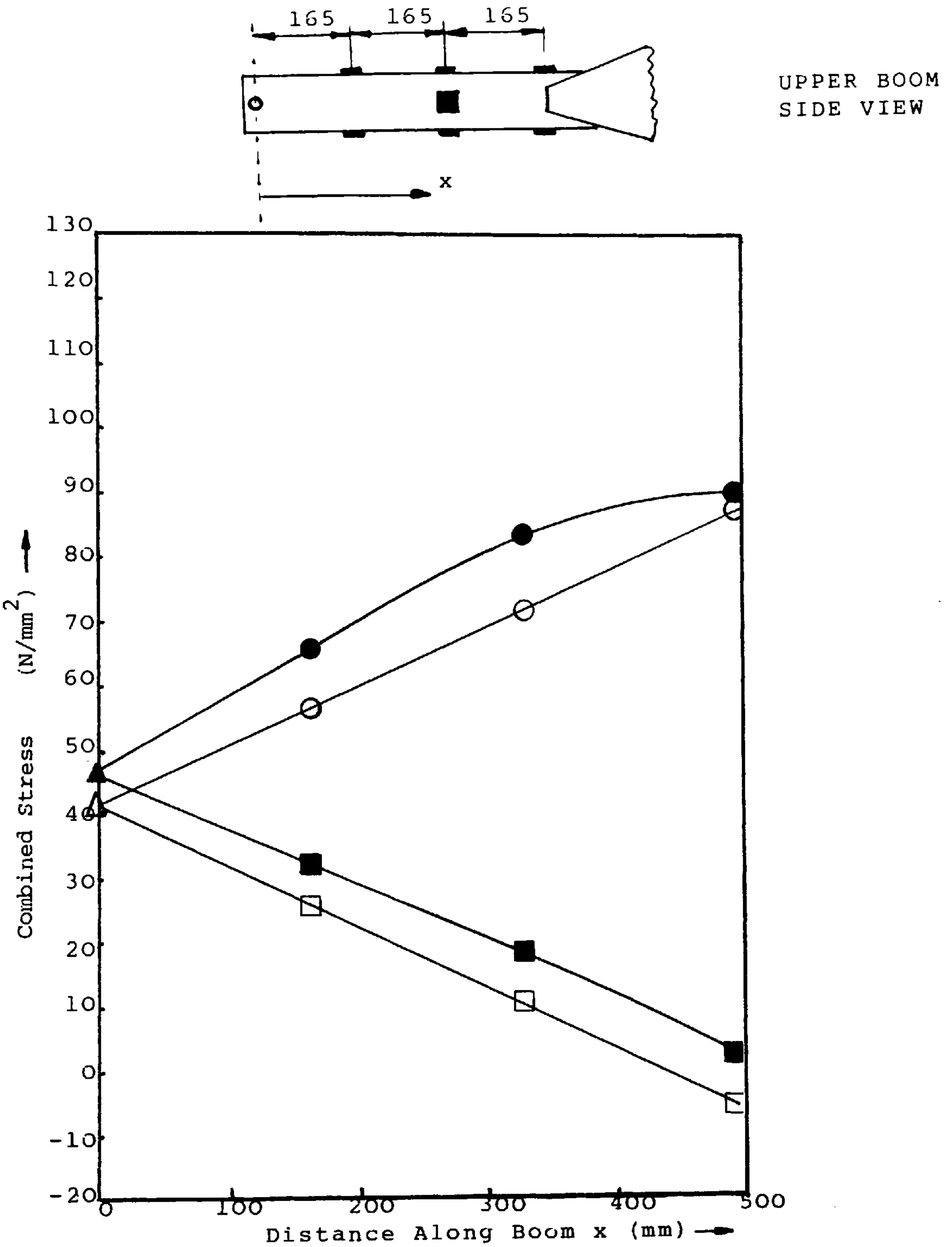
Using the original bucket tooth arrangement it was only possible to load the excavator digging arm against the load cells using the bucket and dipper cylinder; for all configurations the boom cylinder was nearly fully open and for configuration 1 only the dipper cylinder could be used because operation of the bucket cylinder caused the bucket tooth to rotate and foul on the test rig.

To illustrate the result trends Figures 5.1 and 5.2 show graphical plots of the combined stress distribution for configuration 3 (Figure 4.3) using the bucket and dipper cylinder independently.

The figures show that in general the experimental combined stress results are consistently higher than the calculated theoretical results.

The experimental combined stresses on the side surfaces was found to differ. In order to plot the results graphically the average value of combined stress was used and this was assumed to be constant along the length of the upper boom section side surfaces.

The general curvature of the experimental combined stress results is also noticeable towards the boom side plate end of the upper boom (upper boom distance X is between 300 and 500 mm).



KEY

● Experimental Combined Stress on Lower Surface

○ Theoretical Combined Stress on Lower Surface

■ Experimental Combined Stress on Upper Surface

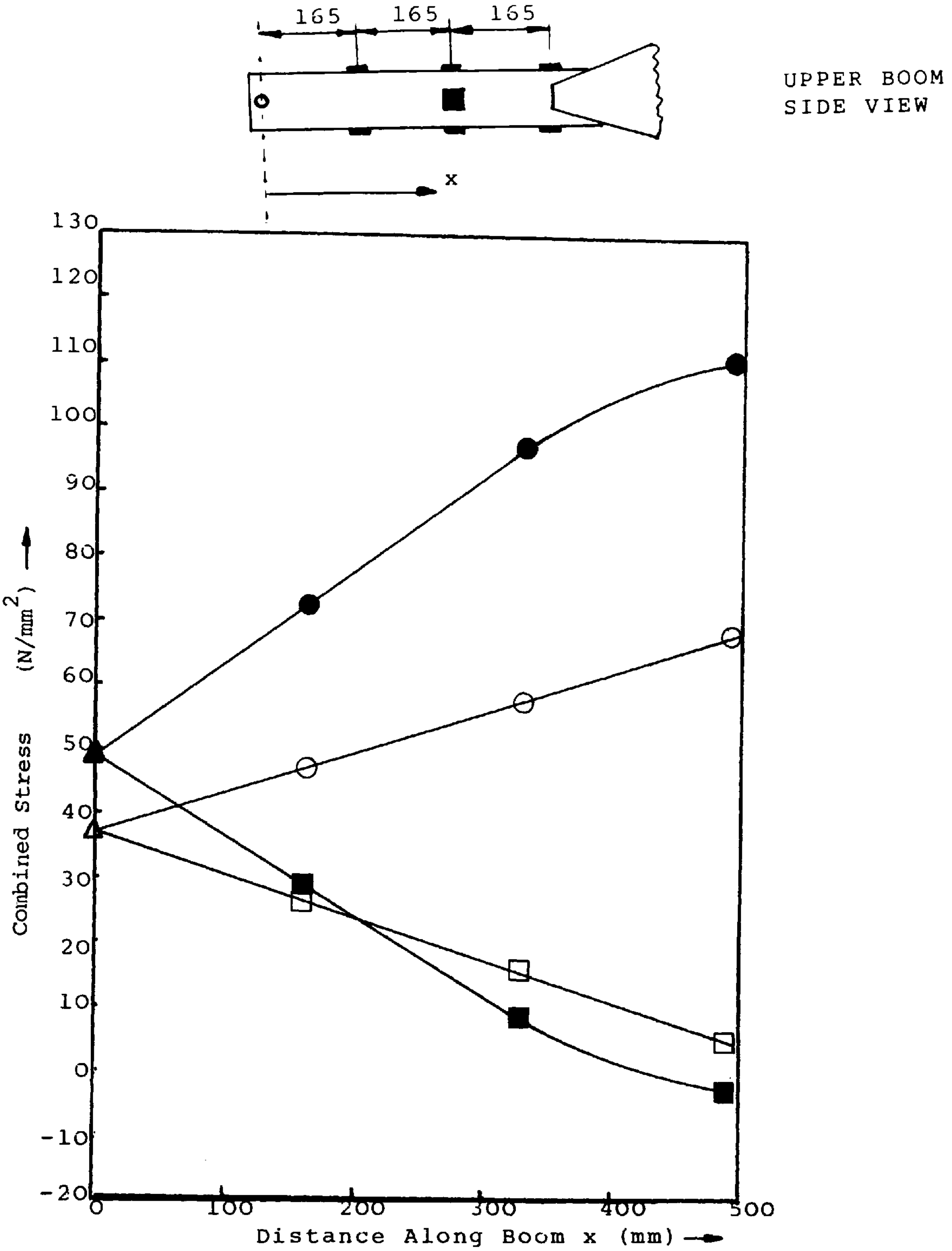
□ Theoretical Combined Stress on Upper Surface

▲ Average Experimental Combined Stress on Side Surface (assumed constant along length)

△ Theoretical Combined Stress on Side Surface

FIGURE 5.1

Experimental and theoretical Combined Stress Distribution, Configuration 3 - Bucket Cylinder (Original Bucket Tooth Arrangement).



KEY

- ● Experimental Combined Stress on Lower Surface
- ○ Theoretical Combined Stress on Lower Surface
- ■ Experimental Combined Stress on Upper Surface
- □ Theoretical Combined Stress on Upper Surface
- ▲ Average Experimental Combined Stress on Side Surface (assumed constant along length)
- △ Theoretical Combined Stress on Side Surface

FIGURE 5.2

Experimental and Theoretical Combined Stress Distribution, Configuration 3 - Dipper Cylinder (Original Bucket Tooth Arrangement).

5.1.2 Theoretical and Experimental Results - Modified Bucket Tooth Arrangement

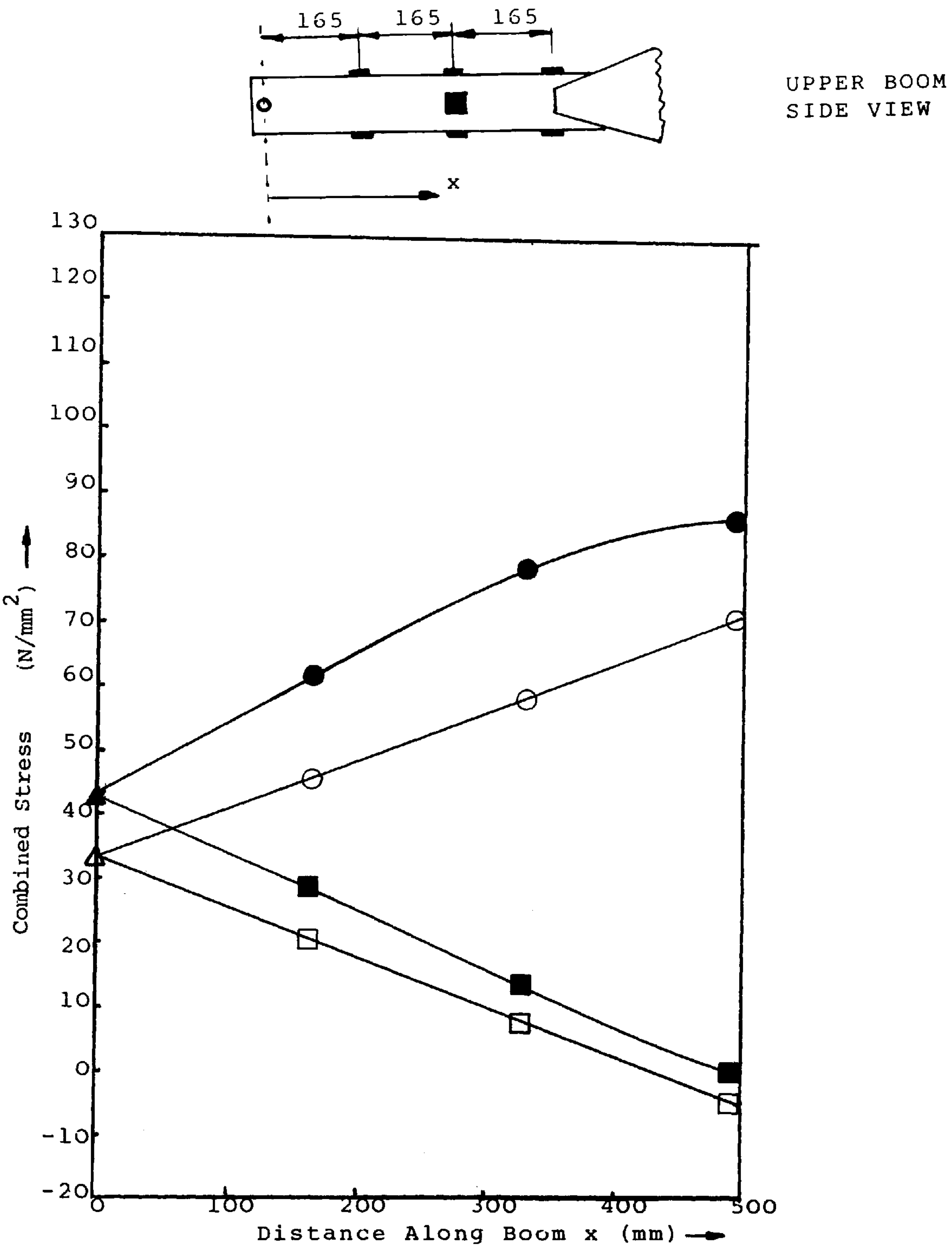
Chapter 4 identified a number of inaccuracies in the measurement of the bucket tooth force using the original bucket tooth arrangement. This section presents the experimental and theoretical results obtained using the modified 'roller type' bucket tooth. With this arrangement it was possible to load the excavator digging arm against the load cells using the bucket and dipper cylinders independently in all three geometric configurations. Again for all configurations the boom cylinder was nearly fully open and could not be used to load the digging arm.

Figures 5.3 and 5.4 illustrate the combined stress result trends for configuration 3 (Figure 4.3) using the bucket and dipper cylinder independently.

These figures also show that in general the modified bucket tooth arrangement produced higher experimental combined stress results than the calculated theoretical results.

The curvature of the experimental combined stress results towards the boom side plate end of the upper boom is again noticeable.

The experimental combined stresses on the side surfaces were found to differ again. The average value of combined stress was again used and assumed to be constant along the length of the upper boom side surfaces.

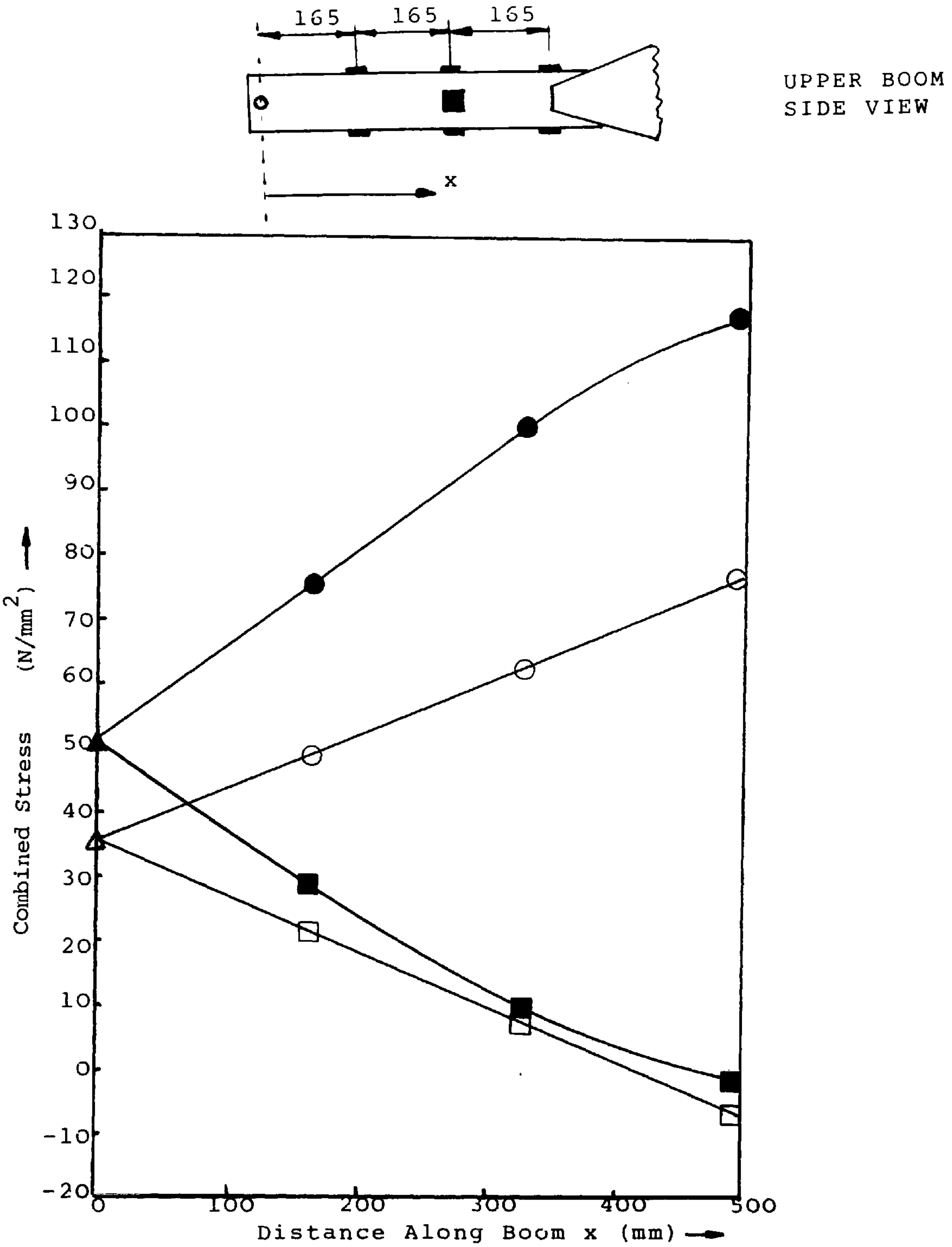


KEY

- ● Experimental Combined Stress on Lower Surface
- ○ Theoretical Combined Stress on Lower Surface
- ■ Experimental Combined Stress on Upper Surface
- □ Theoretical Combined Stress on Upper Surface
- ▲ Average Experimental Combined Stress on Side Surface (assumed constant along length)
- △ Theoretical Combined Stress on Side Surface

FIGURE 5.3

Experimental and Theoretical Combined Stress Distribution, Configuration 3 - Bucket Cylinder (Modified Bucket Tooth Arrangement).



KEY

● Experimental Combined Stress on Lower Surface

○ Theoretical Combined Stress on Lower Surface

■ Experimental Combined Stress on Upper Surface

□ Theoretical Combined Stress on Upper Surface

▲ Average Experimental Combined Stress on Side Surface (assumed constant along length)

△ Theoretical Combined Stress on Side Surface

FIGURE 5.4

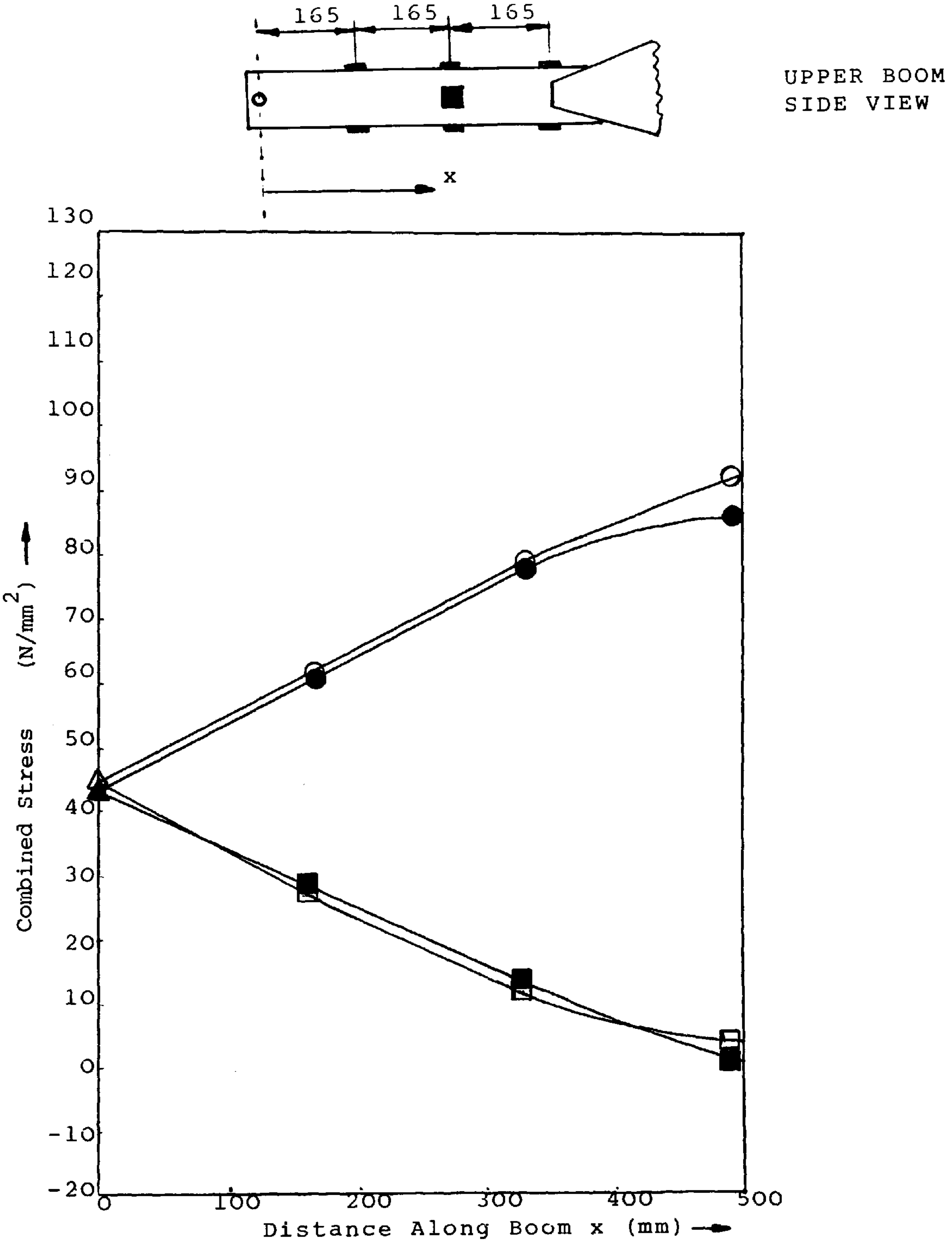
Experimental and Theoretical Combined Stress Distribution, Configuration 3 - Dipper Cylinder (Modified Bucket Tooth Arrangement).

5.1.3 Theoretical and Experimental Results - Torsional Effects

During the initial reduction of experimental data the measured shear stresses were ignored. Further analysis of these results showed that there was an appreciable level of shear stress at many of the gauge locations. Theoretically the torsional shear stress at each location due to a torsional moment Tx is constant along the length of the box section. The actual results showed there to be large variations in shear stress along the length.

The measured shear stresses at each gauge location were used together with equations (3.73) and (3.74) to re-calculate theoretical principal stresses these were calculated for experimental configuration 3 using both the bucket and dipper hydraulic cylinders and the modified bucket tooth to illustrate the effects of torsion. The modified results for the plotted stress distribution are shown in figures 5.5 and 5.6 for the bucket and cylinder operation respectively.

Figures 5.5 and 5.6 show generally, that when torsional effects are considered there is a much better correlation between the experimental and theoretical combined stress results than those shown in Figures 5.3 and 5.4.



KEY

● Experimental Combined Stress on Lower Surface

○ Theoretical Combined Stress on Lower Surface

■ Experimental Combined Stress on Upper Surface

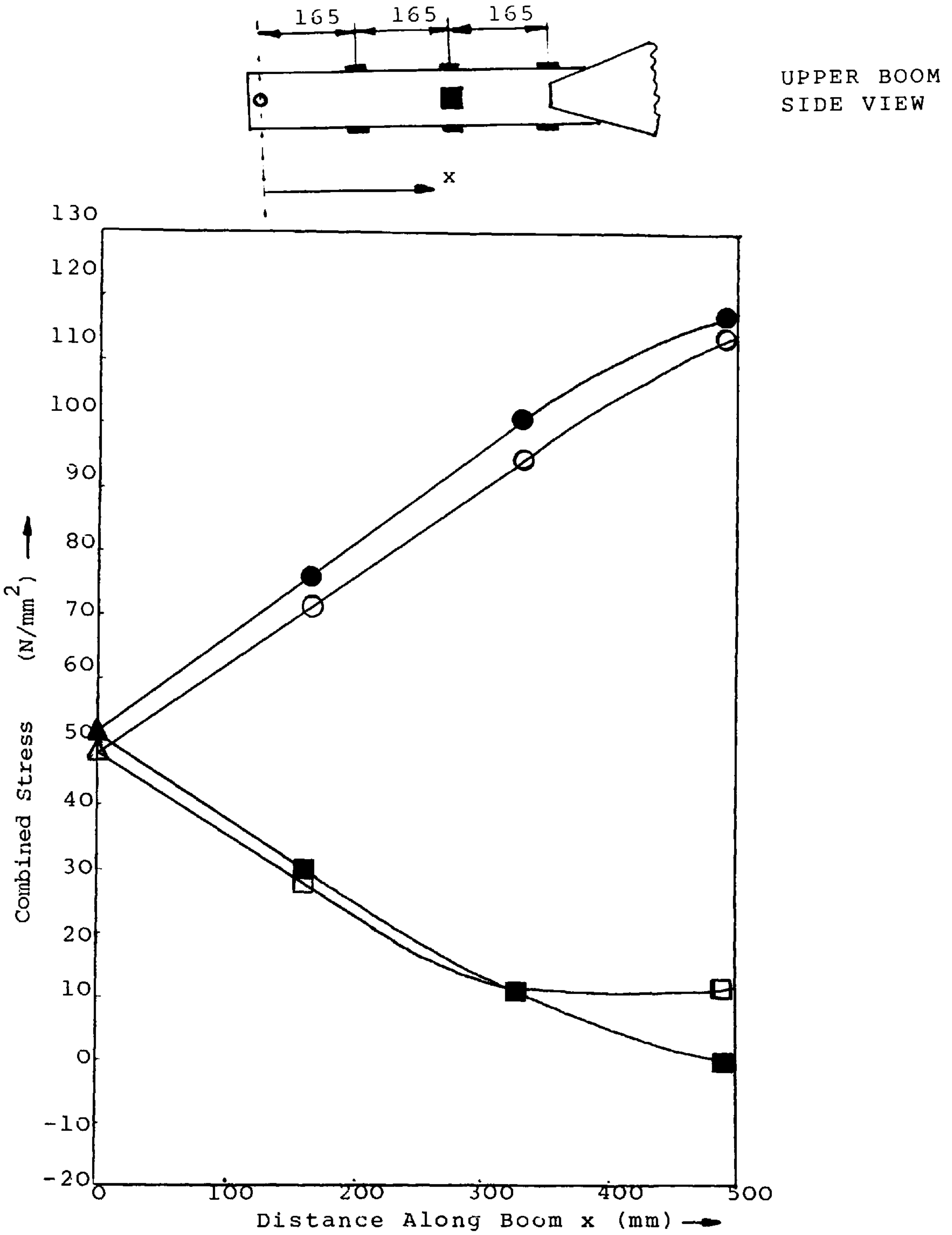
□ Theoretical Combined Stress on Upper Surface

▲ Average Experimental Combined Stress on Side Surface (assumed constant along length)

△ Theoretical Combined Stress on Side Surface

FIGURE 5.5

Experimental and Theoretical Combined Stress Distribution, with Allowance for Torsion, Configuration 3 - Bucket Cylinder (Modified Bucket Tooth Arrangement).



KEY

- ● Experimental Combined Stress on Lower Surface
- ○ Theoretical Combined Stress on Lower Surface
- ■ Experimental Combined Stress on Upper Surface
- □ Theoretical Combined Stress on Upper Surface
- ▲ Average Experimental Combined Stress on Side Surface (assumed constant along length)
- △ Theoretical Combined Stress on Side Surface

FIGURE 5.6

Experimental and Theoretical Combined Stress Distribution, with Allowance for Torsion, Configuration 3 - Dipper Cylinder (Modified Bucket Tooth Arrangement).

5.1.4 Discussion of Structural Analysis Results

The initial assumptions made in 3.1 about the structural model of the digging arm are summarised as follows:

1. Structure of negligible weight.

2. No joint friction.

3. Close fit at joints.

4. No out of plane bending.

5. No torsion on the sections.

The validity of these assumptions is discussed in the section.

The results obtained with the original bucket tooth arrangement showed large discrepancies between the experimental and theoretical combined stress results. Initially the discrepancy was thought to be due to the method used for bucket tooth force measurement.

A modified bucket tooth arrangement was designed in an attempt to improve the accuracy of results. The results obtained with the modified bucket tooth arrangement also showed large discrepancies between the experimental and theoretical combined stress results.

In the light of these results the initial assumptions that were made about the structural model were re-evaluated.

The first assumption made was that the structural weight of the digging arm is negligible. During the experimental tests all the strain gauges were zeroed with the digging arm in the geometric configuration prior to loading with the hydraulic cylinders. Thus any stress recorded by the strain gauges was due to loading using the hydraulic cylinders only and not the structural weight.

The second assumption made was that all joints are frictionless. The lubrication and free rotation of all joints was checked but since the lack of fit allowed free movement of most joints, the effects of friction are minimal.

The third assumption made was that there was a close fit at the joints. During experimentation it was noticed that the freedom of movement of the joints gave rise to considerable mis-alignment of the digging arm.

The fourth assumption made was that no out of plane bending was present because of this stated mis-alignment. The degree of out of plane bending is appreciable when the digging arm is under maximum loading conditions in the loading frame. Further examination of the structural components showed that the sections of the upper boom welded together along the centre line were found to be mis-aligned, consequently the boom hinge pin was not aligned perpendicular to the section sides, but at an angle to them. This resulted in the upper boom and dipper arm being out of alignment.

The fifth assumption made was that there was no torsion present on the sections of the digging arm. The resultant mis-alignment of the components must give rise to some degree of torsion under maximum loading conditions in frame. Observations of the digging arm under loading conditions in the frame showed this to be true.

Re-examination of the experimental results showed there to be appreciable levels of torsional stress at many of the gauge locations. The re-calculated theoretical results obtained with a correction for the experimental measured torsional stress showed a much improved correlation with the experimental combined stress results.

The results prove that the combined effects of lack of fit of the joints, out of plane bending and mis-alignment of welded section give rise to torsion on the upper boom section under normal loading conditions. This can have a significant effect on the overall combined stress levels on the sections of the upper boom.

5.2 Optimisation Results

The design parameters that are used in the design optimisation process of a microexcavator digging arm were discussed in section 3.2. A design optimisation method was developed using the CAD programs described in section 3.3. The design optimisation method is purely theoretical in nature, no account has been taken of the effects of torsion. The design optimisation method is outlined below and the results that were obtained are presented and discussed in the following sections:-

5.2.1 Design Optimisation Method

1. Determine linkage dimensions
   Create linkage dimensions. Data file GEOMXXX
   Using program G360

2. Determine section properties at locations of interest for stress calculations. Create section properties Data File SECTXXX using program H360

3. Determine the range of geometric configurations to be considered and the magnitude and direction of forces at joints due to a unit bucket tooth force using program P360; This creates Force Data File FORCXXX

4. Determine the service line relief value pressure for each hydraulic cylinder. Input these to program D360; this program then calculates the maximum achievable digging force for each geometric configuration and the combined stress and safety factor for a given material at all locations of interest.

   After completing calculations for all configurations, the program outputs the minimum overall safety factor for each location and the average and maximum digging forces for the complete range of geometric configurations.

5. Plot the safety factor versus average and maximum digging force to determine performance.

6. Repeat steps 4 and 5 for a range of service line relief valve pressures.

The linkage dimensions of the Powerfab 360WT Microexcavator were obtained from engineering drawings and a data file GEOM001 was created using Program G360.

The locations of interest for stress calculations were determined through discussion with the Powerfab design engineers (Fig. 5.7). A data file of these section properties SECT001 was created using Program H360.

The range of geometric configurations through which safety factor and digging forces may be analysed is determined by the open and closed length of the hydraulic cylinder and their incremental extensions throughout the range. To simplify the problem and reduce lengthy computational times, each cylinder was assumed to have a stroke of 300mm (12") and each was incremented by steps of 50mm (2") each hydraulic cylinder was considered in 7 incremental positions so that the total number of configurations analysed was 7x7x7 = 343 in total.

Program P360 was used with a unit tooth force to analyse the forces at the joints and create a data files FORC001 containing this data.

In order to assess the current microexcavator digging performance and overall minimum safety, the present service line relief valve settings were input to Program D360. The current settings are:-

Boom cylinder	276 BAR (4000 psi)
Dipper cylinder	276 BAR (4000 psi)
Bucket cylinder	138 BAR (2000 psi)

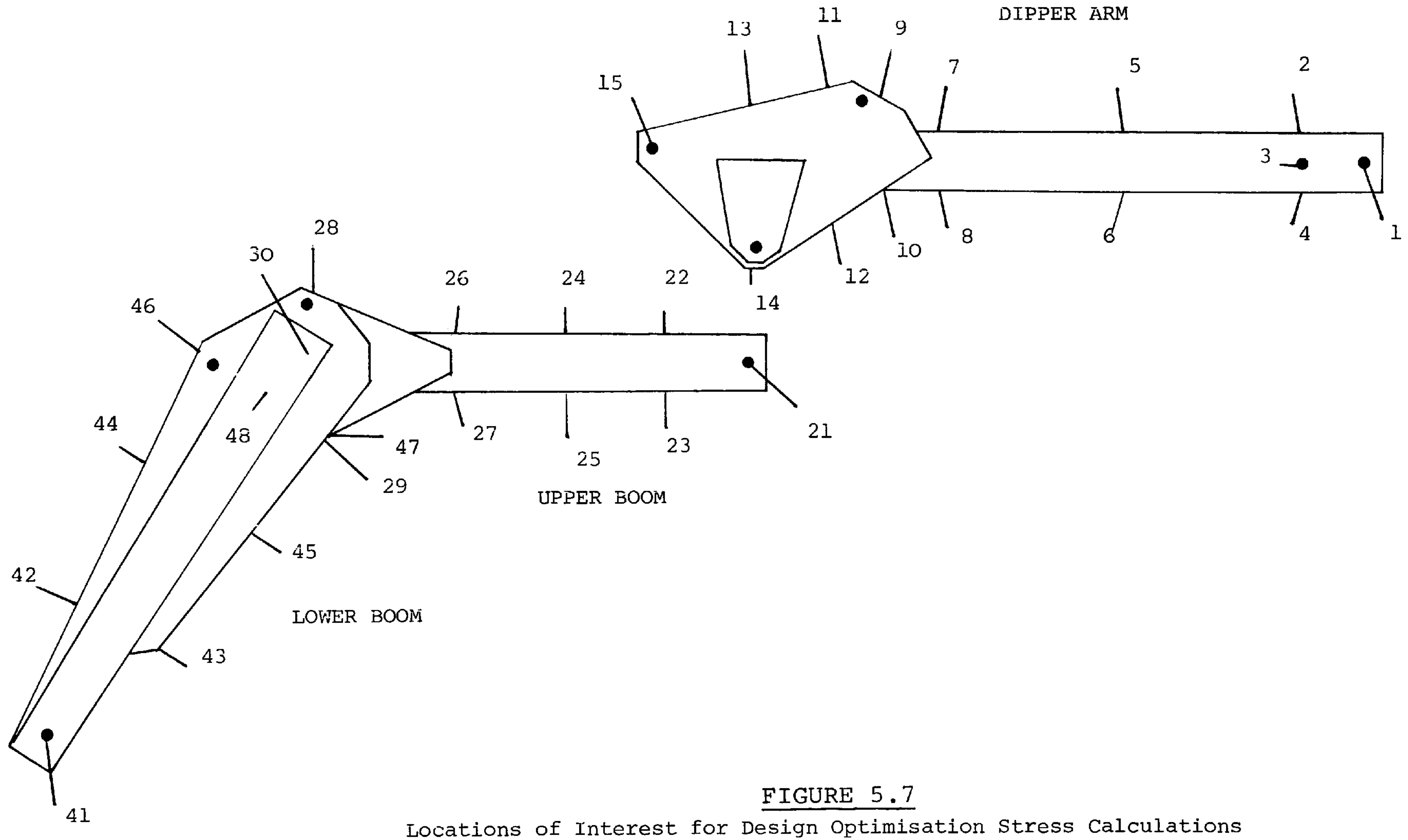


FIGURE 5.7
Locations of Interest for Design Optimisation Stress Calculations

The Table 5.1 shows the program results output for the present Powerfab 360Wt Microexcavator with the above SLRV settings. The maximum and average bucket tooth forces were found to be 12.98kn (1.32 tons) and 8.50 kn (0.87 tons) respectively. The overall minimum safety factor for the complete range of geometric configurations was found to be 1.05 based on a material yield stress of 245 N/mm².

The present Powerfab 360WT digging forces were considered to be adequate for the purpose of design optimisation since users of the equipment in the field have expressed their satisfaction with the present machines performance.

The overall minimum safety factor however was not considered satisfactory, and through discussions with the Powerfab design personnel a desired safety factor of 1.5 was chosen for design optimisation purposes.

Having chosen these design optimisation performance parameters, Program D360 was used again, but this time with a range of SLRV pressures as detailed below:-

Boom cylinder - 138 to 276 BAR at 69 BAR intervals (2000 to 4000 psi at 1000 psi intervals)

Dipper cylinder - as above

Bucket cylinder - 103 to 172 BAR at 34.5 BAR intervals (1500 psi to 2500 psi at 500 psi intervals)

The results obtained are presented and discussed in the following sections.

POWERFAB 360 OPTIMISATION

Hydraulic Cylinder Dimensions and Pressure Settings

Notation	Units
D = Cylinder internal diameter	mm(ins)
d = Cylinder rod diameter	mm(ins)
Ps = Pressure relief valve setting	BAR(psi)

BOOM			DIPPER			BUCKET		
D	d	Ps	D	d	Ps	D	d	Ps
66.68	38.10	275.79	66.68	38.10	275.79	63.50	38.10	137.89
(2.63	1.50	4000.00	2.63	1.50	4000.00	2.50	1.50	2000.00

****

****

Data for Complete Sweep of Geometry

Notation		Units
f	= Percentage no. of failures of ram while max. tooth force is being exerted	
Fav	= Average tooth force for complete sweep of geometry	kN(tons)
Fm	= Max. tooth force achieved for complete sweep of geometry	kN(tons)

BOOM	DIPPER	BUCKET	TOOTH FORCE	
f	f	f	Fav	Fm
39.36	7.29	53.35	8.50	12.98
(39.36	7.29	53.35	0.87	1.32)

****
* * * Minimum Safety Factor = 1.05 * * *
****

For the following conditions
boom length = 745.00 mm
dipper length = 745.00 mm
bucket ram length = 645.00 mm
digging force = 12.98 kN
at 20.00 deg. to tooth

TABLE 5.1

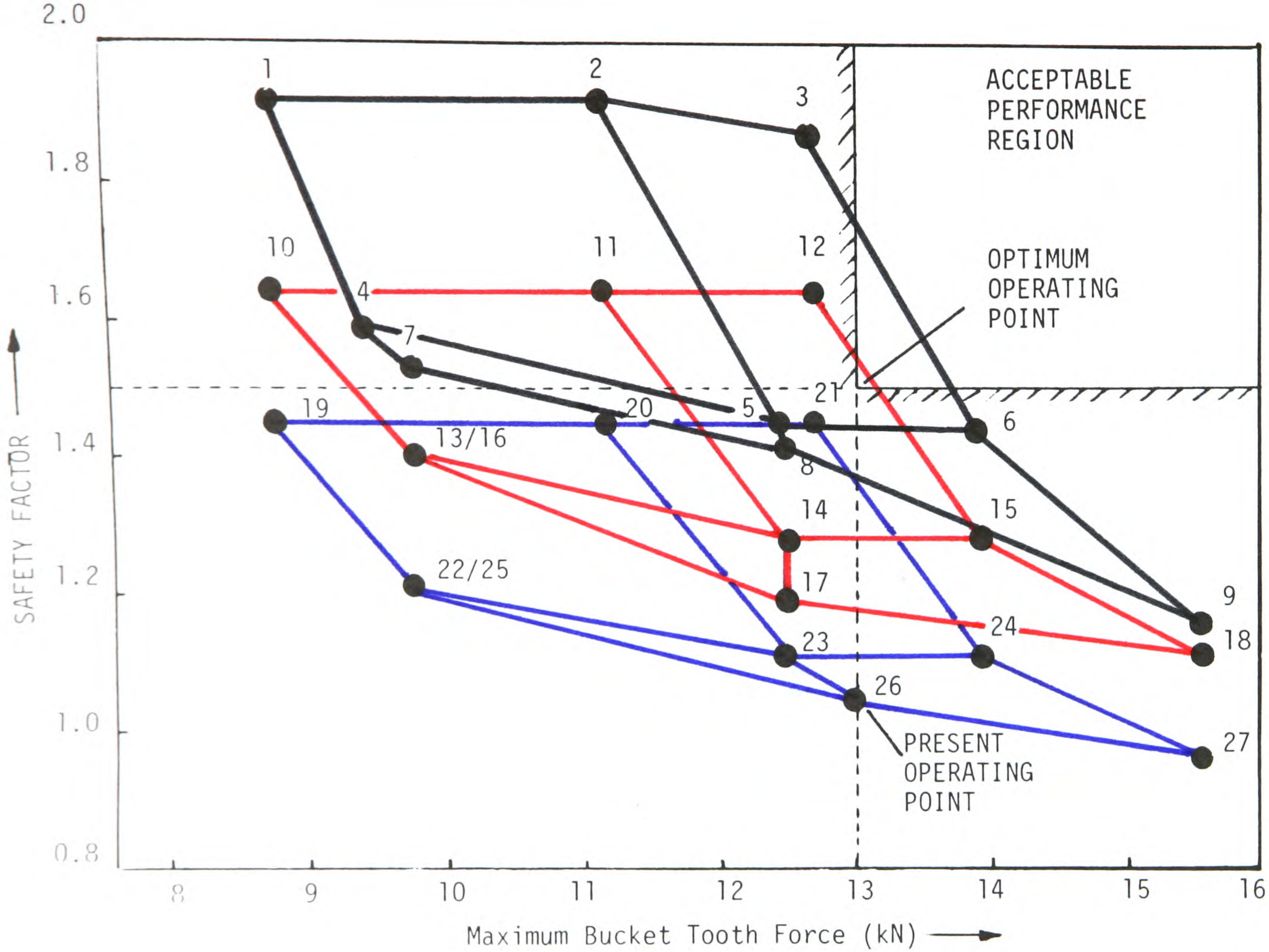
Present POWERFAB 360WT Microxcavator Performance

5.2.2 Discussion of results for - Safety factor versus maximum bucket tooth force

These results are plotted for a range of SLRV settings in Figure 5.8. The horizontal line on the graph represents an acceptable level of safety factor of 1.5 for all locations of interest on the digging arm. The vertical line represents an acceptable level of maximum bucket tooth force of 12.98 kN (1.32 tons), this being the current microexcavator performance. The area bounded by these two lines in the right hand corner of the graph represents the acceptable performance design. each ' operating point' on the graph represents a particular combination of three SLRV settings as shown in the key to the right of the graph. The coloured grids used for clarity represent linear interpolations for operating points of constant boom SLRV setting but changing dipper and bucket SLRV setting. The black grid - represents A Boom SLRV of 138 bar, the red grid represents a Boom SLRV of 207 bar and the blue grid a boom SLRV of 276 bar.

It can clearly be seen that none of the discrete operating points chosen lies within the acceptable performance region. However the linear interpolations between operating points 3 and 6 for boom SLRV 138 bar and operating points 12 and 15 for boom SLRV 207 bar show that some points on these lines fall into the acceptable performance region. Operating point 26 represents the current microexcavator SLRV settings and performance (safety factor 1.05, maximum bucket tooth forces 12.98 kN). Note that of all the discrete points, operating point 21 is closes to the optimum operating points. Each of the grids is similar in shape but moves in position vertically as the boom SLRV pressure setting is reduced. This shows that the safety factor is directly dependent on boom SLRV pressure setting but the maximum bucket tooth force is virtually independent of boom SLRV pressure setting.

Operating Point 26 on Figure 5.8 represents the present operating point for the existing structure (safety factor 1.05, maximum digging force 12.98KN, minimum safety factor at location 27 on the upper boom Figure 5.7).



KEY

OP. POINT	SLRV PRESSURES (BAR) BOOM	DIPPER	BUCKET
1	138.0	138.0	103.5
2	138.0	138.0	138.0
3	138.0	138.0	172.5
4	138.0	207.0	103.5
5	138.0	207.0	138.0
6	138.0	207.0	172.5
7	138.0	276.0	103.5
8	138.0	276.0	138.0
9	138.0	276.0	172.5
10	207.0	138.0	103.5
11	207.0	138.0	138.0
12	207.0	138.0	172.5
13	207.0	207.0	103.5
14	207.0	207.0	138.0
15	207.0	207.0	172.5
16	207.0	276.0	103.5
17	207.0	276.0	138.0
18	207.0	276.0	172.5
19	276.0	138.0	103.5
20	276.0	138.0	138.0
21	276.0	138.0	172.5
22	276.0	207.0	103.5
23	276.0	207.0	138.0
24	276.0	207.0	172.5
25	276.0	276.0	103.5
26	276.0	276.0	138.0
27	276.0	276.0	172.5

FIGURE 5.8

Safety Factor versus Maximum Bucket Tooth Force for a Range of Service Line Relief Valve Settings

The locations at which the minimum safety factor occurs for the complete range of SLRV settings are 7 and 8 on the dipper arm and 27 on the upper boom section).

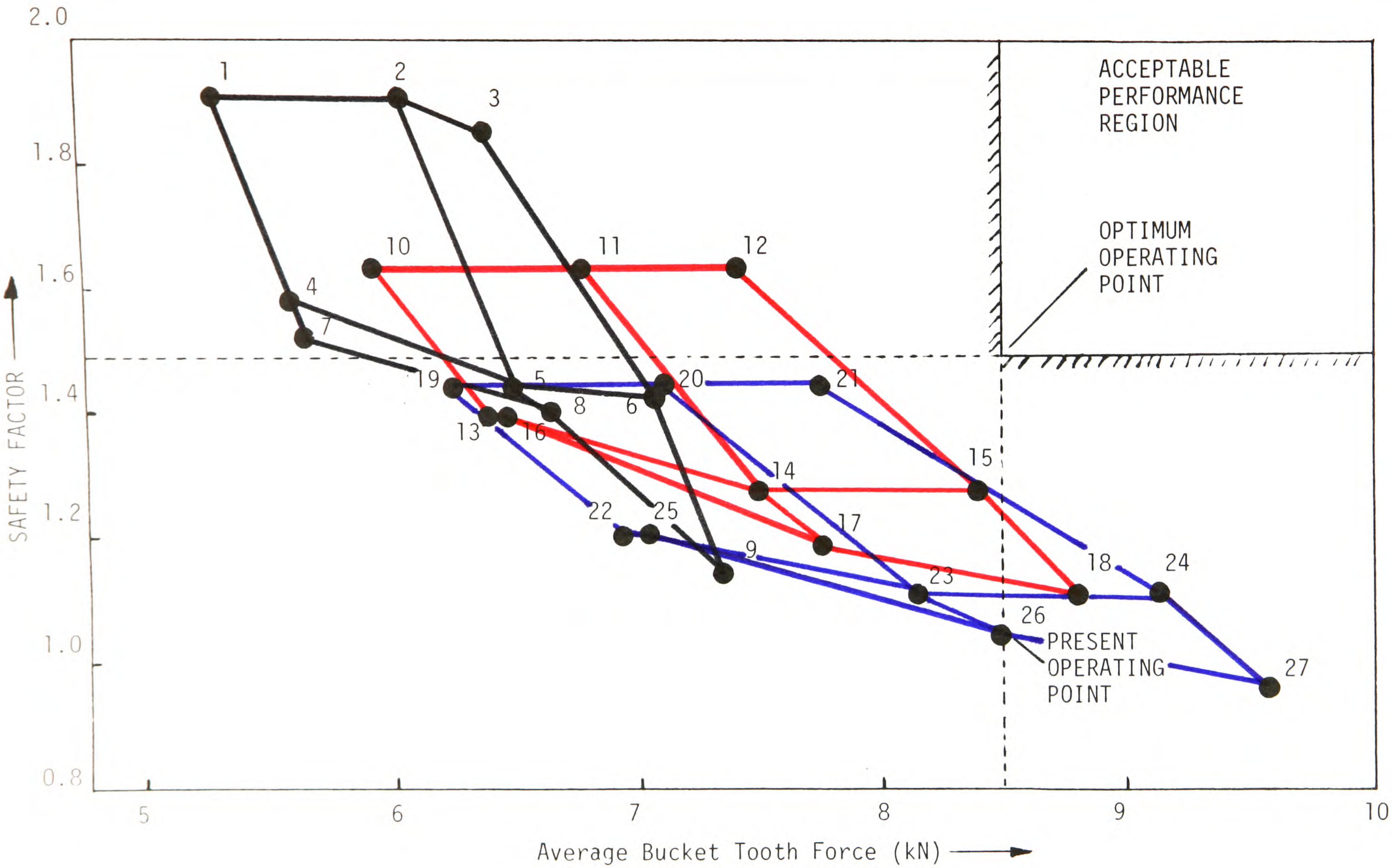
5.2.3 Discussion of Results for - Safety factor versus average bucket tooth force

These results are plotted for a range of SLRV settings and are shown in Figure 5.9. The horizontal line of the graph represents an acceptable level of safety factor of 1.5 and the vertical line represents an acceptable level of average bucket tooth force of 8.50 kN (0.87 tons), this being the present microexcavator performance. The acceptable performance region appears in the right hand corner of the graph and coloured grids are used for clarity as in Figure 5.8.

It can clearly be seen that none of the discrete operating points chosen lies within the acceptable performance region. Also this time the linear interpolations between operating points show that no points fall into the acceptable performance region. However some points on the line joining operating points 12 and 15 are in close proximity to the optimum operating point. Operating point 26 again represents the current microexcavator SLRV settings and performance (safety factor 1.05), average bucket tooth force 8.50 kN). Note that of all the discrete points, operating point 21 is closest to the optimum operating point. This time the shape of the grids change dramatically with SLRV pressure settings.

5.2.4 Maximum bucket tooth force versus SLRV pressure setting

These results are plotted in three dimensional form for clarity and are shown in figure 5.10. The x-axis of the plot represents the variation of bucket SLRV pressure, the y-axis represents the variation of dipper SLRV pressure and the z-axis represents the variation of boom SLRV pressure. Each node of the 'cube' represents one combination of all three SLRV pressures and is numbered 1 to 27, the values for maximum bucket tooth force appear at each node below the node number.



KEY

OP. POINT	SLRV PRESSURES (BAR) BOOM	DIPPER	BUCKET
1	138.0	138.0	103.5
2	138.0	138.0	138.0
3	138.0	138.0	172.5
4	138.0	207.0	103.5
5	138.0	207.0	138.0
6	138.0	207.0	172.5
7	138.0	276.0	103.5
8	138.0	276.0	138.0
9	138.0	276.0	172.5
10	207.0	138.0	103.5
11	207.0	138.0	138.0
12	207.0	138.0	172.5
13	207.0	207.0	103.5
14	207.0	207.0	138.0
15	207.0	207.0	172.5
16	207.0	276.0	103.5
17	207.0	276.0	138.0
18	207.0	276.0	172.5
19	276.0	138.0	103.5
20	276.0	138.0	138.0
21	276.0	138.0	172.5
22	276.0	207.0	103.5
23	276.0	207.0	138.0
24	276.0	207.0	172.5
25	276.0	276.0	103.5
26	276.0	276.0	138.0
27	276.0	276.0	172.5

FIGURE 5.9

Safety Factor versus Average Bucket Tooth Force for a Range of Service Line Relief Valve Settings

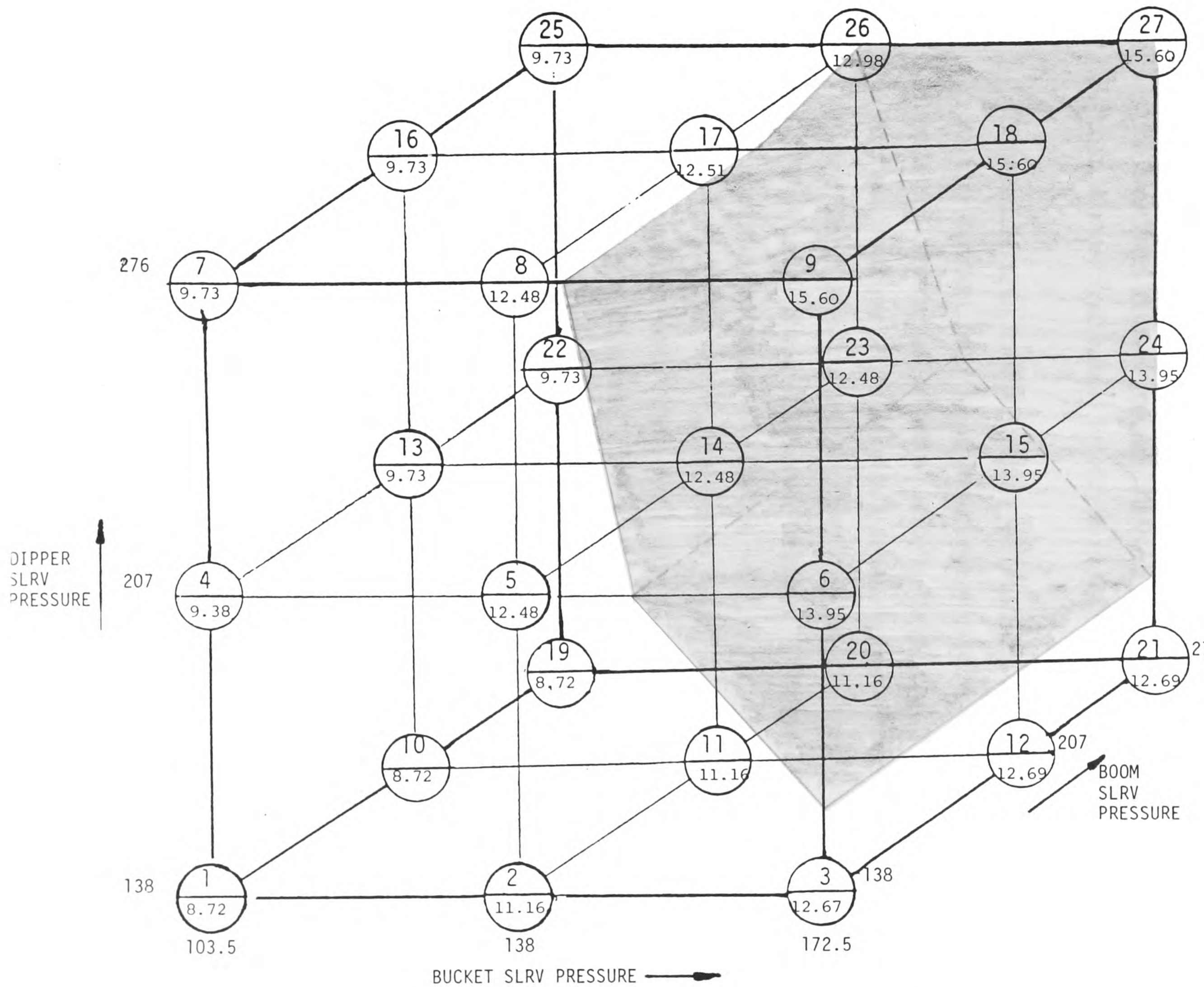


FIGURE 5.10

Variation of Maximum Bucket Tooth Force (kN) with SLRV Pressure Setting (BAR)

From the plot it can clearly be seen that the maximum bucket tooth force remains virtually constant when the boom SLRV pressure changes. The maximum bucket tooth force is therefore virtually independent of boom SLRV pressure. As the dipper SLRV pressure increases the maximum bucket tooth forces also increases. The increases are small for lower bucket SLRV pressure settings, becoming greater for higher values of bucket SLRV prssure settings. As the bucket SLRV pressure increases the maximum bucket tooth force also increases by greater amounts than those indicated by dipper SLRV pressures.

Generally the lowest values of maximum bucket tooth force appear in the bottom left hand near corner of the cube in the region of node 1, where all three SLRV's have their lowest values. The highest values of maximum bucket tooth force appear in the top right hand far corner in the region of node 27 where all three SLRV's have their highest values. Using linear interpolation the points between the nodes with a maximum bucket tooth force value of 12.98 kN (1.32 tons) were found (see section 5.2.1); these points were then joined and the volume formed shaded as shown in the diagram. This shaded area represents combinations of SLRV pressure settings that satisfy the acceptable maximum bucket tooth force criteria.

Nodes 6, 9, 15, 18, 24, 26 and 27 all yield maximum bucket tooth values at or above the acceptable maximum bucket tooth force value of 12.98 kN (1.32 tons), and all lie within the shaded area.

5.2.5 Average bucket tooth force versus SLRV pressure setting

These results are also plotted in three dimensional form for clarity and are shown in Figure 5.11.

From the plot it can clearly be seen that the average bucket tooth force is directly proportional to all three SLRV pressure settings. Generally the lowest values of average bucket tooth force appear in the bottom left hand corner of the cube in the region of node 1 and the highest values in the top right hand far corner in the region of node 27. Linear interpolation was

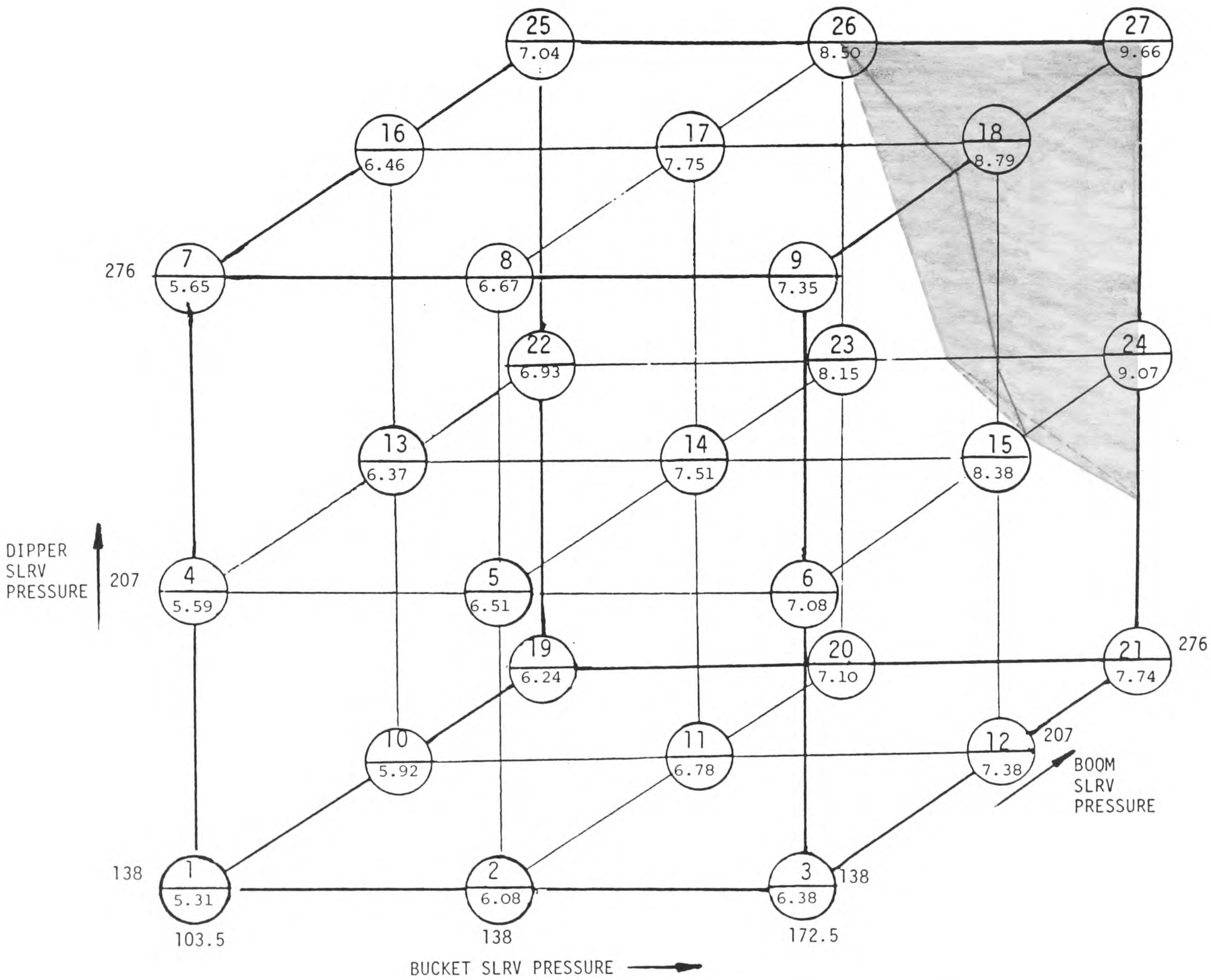


FIGURE 5.11

Variation of Average Bucket Tooth Force (kN) with SLRV Pressure Setting (BAR) again used to determine the shaded area to satisfy the acceptable average bucket tooth force criteria. Nodes 18, 24, 26 and 27 all yield average bucket tooth forces at or above the acceptable average bucket tooth force value of 8.59 kN (0.87 tons) and all lie within the shaded area.

5.2.6 Safety factor versus SLRV pressure setting

These results are also plotted in three dimensional form for clarity and are shown in Figure 5.12.

From the plot it can be seen that this time safety factor is inversely proportional to all three SLRV pressure settings. Generally the lowest values of safety factor appear in the top right hand far corner of the cube in the region of node 27 and the highest values appear in the bottom left hand near corner in the region of node 1. Linear interpolation was used again to determine the shaded area to satisfy the acceptable safety factor criteria. Nodes 1, 2, 3, 4, 7, 10, 11 and 12 all yield safety factors at or above the acceptable safety factor value of 1.5 and all lie within the shaded area.

5.2.7 Optimisation results - Conclusion

The design optimisation plots demonstrate that useful design optimisation data can be obtained using discrete points (selected combinations of SLRV pressure settings). The following general trends were deduced from the optimisation plots.

1. Maximum bucket tooth force is directly proportional primarily to the bucket SLRV pressure setting and secondly to the dipper SLRV pressure setting but independent of the boom SLRV pressure setting.

2. Average digging force is directly proportional to primarily the bucket SLRV pressure setting, secondly the dipper SLRV pressure setting and thirdly the boom SLRV pressure setting.

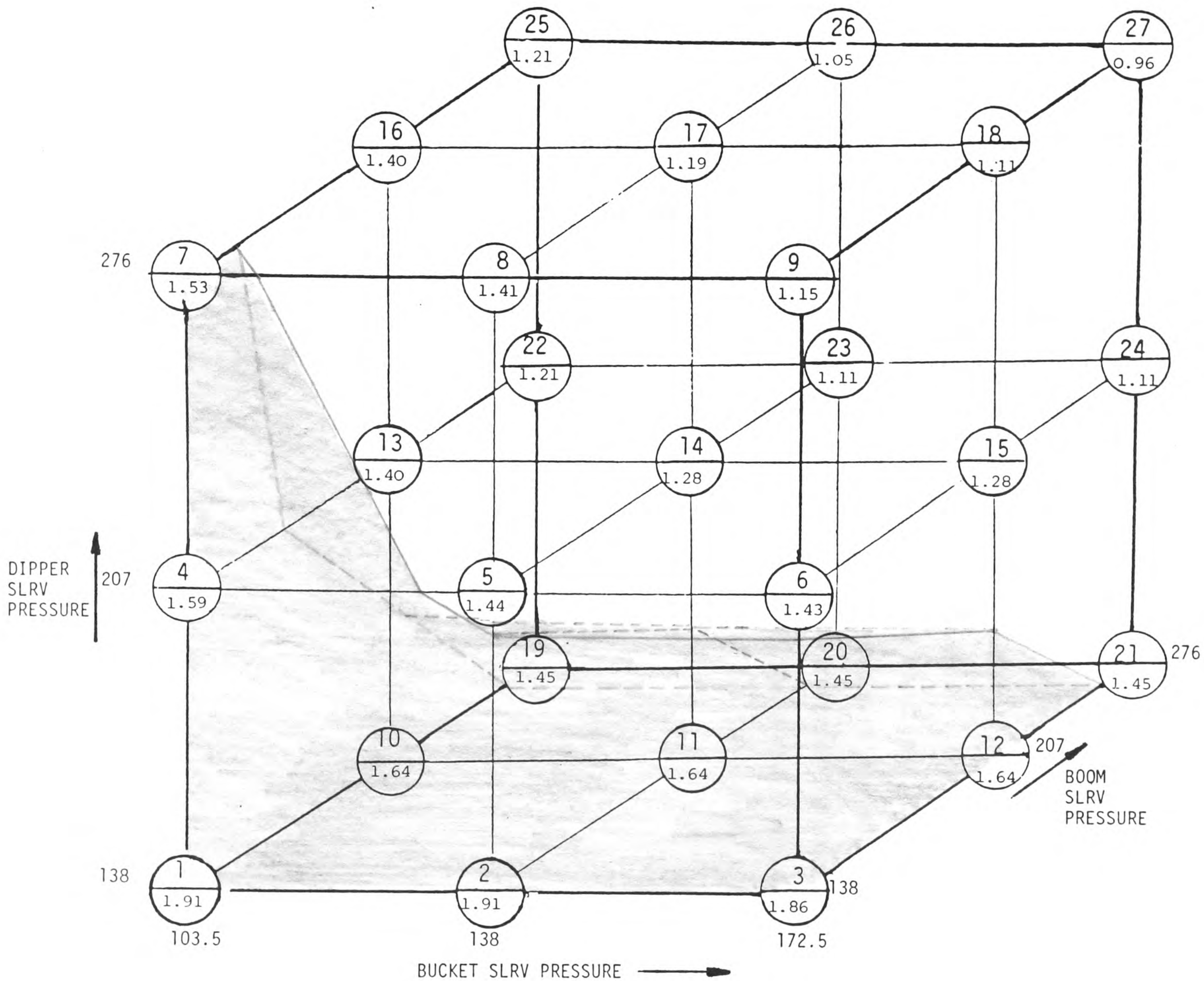


FIGURE 5.12

Variation of Safety Factor with SLRV Pressure Setting (BAR)

3. Safety factor is inversely proportional to the bucket, dipper and boom SLRV pressure settings.

The results show that it is not possible to achieve an acceptable maximum bucket tooth force, average bucket tooth force and safety factor with one combination of the three SLRV pressure settings. However, of the discrete points chosen in the analysis operating point 21 comes closest to satisfying the design optimisation criteria. Using linear interpolation it has been shown that it is possible to achieve both an acceptable maximum bucket tooth force and safety factor but the average bucket tooth force is decreased. The present Powerfab microexcavator performance shows that an increase in safety factor of approximately 50% is necessary to achieve an overall acceptable safety factor of 1.5. In order to achieve this the boom and dipper SLRV pressure settings should be reduced and the bucket SLRV pressure setting should be increased.

CHAPTER SIX

6. CONCLUSIONS AND FUTURE WORK

This chapter discusses the conclusion deduced from the study and describes the future work that could be undertaken.

6.1 Design Considerations

A detailed analysis of the design parameters involved in the design of a microexcavator digging arm under static or quasi-static loading conditions has been successfully completed. The hydraulic system, structure, digging and lifting stability have all been discussed and a better understanding of the design process has resulted.

6.2 Computer Modelling

Fundamental mechanics and structural theory have been used to develop a structural model of the Powerfab 360 microexcavator digging arm and this has been incorporated into a design optimisation method.

6.2.1 Structural modelling

The developed structural model can be used to analyse the forces at the joints and the stresses at locations of interest on the digging arm structure for any geometric configuration and applied bucket tooth force.

6.2.2 Design optimisation

An awareness of the design parameters in the design optimisation process has been gained. A design optimisation method has been developed based on the fundamental computer model and basic hydraulic theory. This optimisation method can be applied to existing and future designs. Initial design optimisation analysis of the existing Powerfab 360 has shown that structural changes are necessary to achieve an acceptable safety factor.

6.2.3 Computer Aided Design (CAD) software

A complete integrated CAD software package has been developed for the design analysis of the Powerfab 360WT microexcavator digging arm. The package allows the user to analyse the effects of design changes to both the digging arm structure and hydraulic system to achieve optimum performance.

6.3 Experimental Tests

A complete data acquisition and analysis system has been developed for the structural analysis of a microexcavator. A structural testing method has also been developed and tests carried out have shown that the fundamental computer model can provide useful design data to an acceptable degree of accuracy when a suitable correction factor for the effects of torsion is employed.

6.4 Future Work

Further structural testing could be carried out on other components of the digging arm to assess the accuracy of the computer modelling techniques.

Further design optimisation calculations could be made using smaller SLRV pressure increments, this would give more points to use in the design optimisation plots and hence a more accurate analysis of trends.

The computer modelling of the microexcavator has shown that the working safety factor of the present Powerfab 360WT digging arm is as low as 1.05 when the digging arm is loaded and the digging platform is fixed to the ground. In 95% of the usual working conditions the digging arm is not fixed, but stands on its four outriggers on the ground. The 'instability' of this arrangement when large bucket tooth forces are applied acts as a safety limit. Further analysis of the effects of static stability on the achievable digging forces would be most useful.

The behaviour of the digging arm has only been considered under static or quasi-static loading conditions. The method could be extended to analyse the effects of dynamic forces on the digging arm structure and hydraulic system.

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APPENDIX 1

POWERFAB/POLYTECHNIC OF WALES

Teaching Company Programme

Software Project Documentation Standards

M. A. Bromfield
October 1986

CONTENTS

APPENDICES

1. INTRODUCTION

One of the most important aspects of any software based project is the documentation of the computer software. This must be carried out both for the user and the programmer if the project is to be completed and supported successfully throughout. The user requires documentation to allow him to fully understand the capabilities of the software and how to use them. The documentation should allow first time users to use the programs (learning by example) and understand how to use them interactively. This should eliminate the problem of only one member of staff being fully conversant with the software package.
The users documentation should allow the user/operator to identify when errors have been made and how to correct them and successfully use the software.
The computer programmer however requires a different set of documentation. The programmers documentation should allow the programmer to translate the customers requirements into software terms. Modifications undoubtedly have to be made throughout the life of the software and this task of software support is made much easier with the aid of comprehensive formal software documentation.
Both types of documentation and the different levels of documentation are illustrated in Table 1.

	USER	PROGRAMMER
level 1	General system description	Functional spec.
level 2	General system user guide General system operator guide	Detailed Des. spec.
level 3	Program user guides Program operator guides	Program spec.
level 4	-----------	Module spec.

Table 1. Software Project Documentation

2. USER DOCUMENTATION

This section describes the formal documentaion required by the user in the Powerfab/Polytechnic of Wales Teaching Company Programme. There are generally three levels of documentation required as detailed below:-

level 1	General system description
level 2	General system user guide General system operator guide
level 3	Program user guides Program operator guides

The relationship of this documentation is illustrated in Table 2.

```
                     General System Description
                               !
                               !
                               !
                  ---------------------------------
                  !                               !
                  !                               !
                  !                               !
   General System User Guide     General System Operator Guide
                  !                               !
                  !                               !
                  !                               !
          ------------------                --------------
          !                !                !            !
          !                !                !            !
          !                !                !            !
Program User Guide             Program Operator Guide
```

Table 2. User Documentation Hierarchy

2.1 General System Description

This section should describe in general terms the overall functions of the complete software package. It is formed from the customers specification in conjunction with the consultant. It should enable the consultant to clearly define the functional specification for the programmers use.

2.2 General User Guide

The general user guide is a brief guide to the use of the complete software package. It provides a general description of each program and the method by which programs are loaded and run. It forms the link between the general system description and the individual program user guides.

2.3 General Operator Guide

The general operator guide should outline the general procedure for correct operation of the system. This should include error identification and methods for error correcton and re-running of the program.

2.4 Program User Guide

Each program requires a detailed program user guide. It should fully describe how to use the program, what screen displays are present, how to respond to prompts etc. A screen display flow chart is quite useful here to deterine the flow of the program. Each user guide should also include a number of examples to enable the first time operator to get used to the program and understand it better.

2.5 Program Operator Guide

The program operator guide should include a list of error codes and descriptions associated with the program. This enables the user to identify where the program went wrong and how to correct it. Procedures for re-running etc. should also be included.

3. PROGRAMMERS DOCUMENTATION

This document describes the formal documentation standards used for the Powerfab/Polytechnic of Wales Teaching Company Programme. It defines the content and purpose of the following specifications for computer programmer's use:-

Level 1	Functional Specification (FS).
Level 2	Detailed Design Specification (DDS). File Specification.
Level 3	Program Specification (PS).
Level 4	Module Specification (MS).

Table 3 shows the relationship of the specifications noted above.
All software documents are intended to be mutually exclusive i.e. specific information is described only in one document.
Sometimes for clarity additional information is referenced in the text.

```
                    FUNCTIONAL SPECIFICATION
                             (FS)
                              !
                              !
                              !
         ---------------------------------------
         !              !              !              !
         !              !              !              !
DETAIL DESIGN          DDS            DDS      GENERAL SOFTWARE
SPECIFICATION                                     SPECIFICATION
    (DDS)                                             (GSS)
      !
      !
      !
  ----------------------------------
  !                !                !
  !                !                !
 PS             PROGRAM             PS
             SPECIFICATION
                 (PS)
                   !
                   !
                   !
       ------------------------
       !           !          !
      MS        MODULE        MS
             SPECIFICATION
```

Table 3. Software Documentation Hierarchy

3.1 Functional Specification (FS)

The functional specification describes briefly the overall functions of the complete software package. It allows all the functional information discussed with the customer to be brought together for clarification. It is usually developed from the customers specification.
It should provide an up to date and definitive description of the system to be used both as a basis for system design and to establish the objectives for system acceptance tests by ther customer.
There is no set layout since this depends upon the area of the system being described.
It should not describe how the functions are to be implemented in software; this information should appear at level 2, the DDS level.
The FS should reference all lower level DDS documents that are relevant.

3.2 Detail Design Specification (DDS)

The DDS is used to expand the functional information in the FS and translate it into software terms. It provides a link between the FS's and the individual program specifications.

As with the FS's the form of the DDS will depend upon the functions being described. For translation of the functions into software terms the following information will be provided:-

1. Communication diagrams showing the data paths between programs (common access data files).

2. Control description or diagrams to show how programs are run and the logical relationship between them.

3. Design constraints - Computer system requirements, program size limits etc.

4. Summary of the functions performed by each program.

5. Data file usage; it is important to describe which programs create/modify/delete data files.

The DDS will be used both as an aid to overall system understanding and as a reference document during program design. The functional description should come first, followed by software description with most diagrams and tables at the end.

3.3 Program Specification (PS)

The program (or task) is the smallest independent scheduable piece of software. The program specification includes the following:-

1. Identification

2. Function

3. Enviroment

4. External References

5. Structure

6. Module Interfaces

7. Supporting Documentation

8. Test Plan

A Program Specification blank should be used each time (Appendix 1.1)

3.3.1 Identification

The program identification appears at the top of the specification.

3.3.2 Function

The function of the program should be described briefly here.

3.3.3 Enviroment

The operating enviroment of the program should be decribed here; this should include the minimum system requirements.

3.3.4 External References

This section describes all data files accessed and any general purpose subroutines used.

3.3.5 Structure

This section should include a routine hierarchy diagram and a routine function summary.

3.3.6 Module Interface

This section describes the module parameters used and any change to external data files if necessary.

3.3.7 Supporting Documnetation

This should include any references to supporting documentation that may assist the understanding of the operation.

3.3.8 Test Plan

This describes how the program should be tested upon completion.

3.4 Module Specification

The module specification is the document from which the computer code is produced by the programmers. It contains the following sections:-

1. Identification
2. Function
3. External References
4. Processing Logic
5. Program Design Language/Flow Charts
6. Supporting Documentation
7. Test Specification
8. Reported errors

The module header blank should be used at all times.(Appendix 1.2)

3.4.1 Identification

The module identification again appears at the top of the module specification.

3.4.2 Function

The function of the routine is briefly described here.

3.4.3 External References

This describes all accessing of other routines, external data files and operating system facilities.

3.4.4 Processing Logic

This contains a description of the processing written in narrative form which must conform to the flow chart and PDL shown in the next section.

3.4.5 PDL/Flow Charts

A full description of the program using PDL or a flow chart should be used. Appendix1.3contains a brief description of PDL, generally PDL will be used.

3.4.6 Supporting Documentation

Any other documents which may assist in the understanding of the module must be detailed here.

3.4.7 Test Specification

This specifies the paths through the module to be tested.

3.4.8 Reported Errors

This section details any reported errors; it should note the error number and corresponding description.

3.5 File Specification

This section describes all the files used by the system; the following section headings should be used:

1. Identifiaction
2. Purpose
3. Physical Attributes
   - Media
   - Approx size
4. Structure
5. Access and update
6. Data sheets for each data record

APPENDIX 1.1

PROGRAM SPECIFICATION

Project :
Title :
Author(s) :

Version :
Created :
Revised :

Ammendments :

PROGRAM :

----

1. Name:

2. Function

3. Environment

4. External References

Routines called

Files accessed

5. Structure

```
                    T360M00
                       !
                       !
                       !
         ------------------------------
         !                !           !
         !                !           !
         !                !           !
      T360M01          T360M02     T360M03
                          !
                          !
                          !
                 --------------
                 !            !
                 !            !
                 !            !
             T360M021     T360M022
```

Hierarchial Structure of the Program

6. Module Interface

T360M00

T360M01

T360M02

T360M03

T360M021

T360M022

7. Supporting Documentation

8. Test Plan

APPENDIX 1.2

PROCEDURE SPECIFICATION

Project :
Title :
Author(s) :

Version :
Created :
Revised :

Ammendments :

PROGRAM :
PROCEDURE :

---

1. Name:

2. Function

3. External References

   Routines called

   Files accessed

4. Processing Logic

5. Program Design Language

6. Supporting Documentation

7. Test Plan

8. Reported Errors

APPENDIX 1.3

PROGRAM DESIGN LANGUAGE

The Program Design Language (PDL) presented here is a tool for designing programs in detail prior to coding.

The PDL is a form of pseudocode and has the following characteristics:-

1. Notation is used to state program logic and function in an easy to read, top to bottom manner.

2. It is not a compilable language.

3. It is an informal method of expressing structured programming logic.

4. It is similar to programming languages without being bound by formal syntactical rules.

5. Conventions are incorporated to aid the visual perception of the logic.

6. The primary purpose is to enable ideas to be expressed in simple English.

7. The language permits concentration on logical solutions to problems, rather than the form and constraints within which the solutions must be coded.

8. The language replaces flow charting and produces complete technical program logic documentation.

9. Program design is expressed readably and can be converted easily to executable code.

An example of PDL is given at the end of this appendix.

Program Design Language - example

```
START
     setup graphics mode and data format
     PROC_T360M01  calc geometry data
     PROC_T360M02  calc reactions
     PROC_T360M03  calc bucket forces
     PROC_T360M04  calc bucket linkage forces
     PROC_T360M05  calc boom forces
     PROC_T360M06  calc forces balance
     write data to file?
     IF answer YES
     THEN PROC_T360M07 output data to file
     calculate stresses?
     IF answer YES
     THEN INPUT program name and CHAIN program
END
```

APPENDIX 1.4

POWERFAB/POLYTECNIC OF WALES

Teaching Company Programme

BBC BASIC Programming Standards

M. A. Bromfield

October 1986

This document outlines the standards used for structured programming using the BBC BASIC programming language.

1. Programming Structure

A BASIC program consists of a Control Routine (Module M00) and a number of procedures or modules. Procedures must have only one entry point and must exit via a standard ENDPROC.
Functions are rarely used and only modify local data and any machine dependent features must be explained using clear and precise REM statements.

2. Fundammental Concepts

There is no place in software where 'clever' coding offers great advantage. Consequently all computer code should be implemented in as simple a manner as possible. Code must correspond to the PDL or flow chart description. All symbol names should be meaningful.

3. Statement Layout

Comments in BBC BASIC are implemented using REM statements; each block of REM statements should have a clear line (line number followed by one space) before and after it.
Within the program the control routine should be numbered 0 - 9999 in steps of 10 where applicable.
All procedures will adopt a five figure number commencing at 10000 in blocks of 1000 in steps of 10 where possible.
Line numbers appear right justified in columns 1 to 5 (0 to 99999). Columns 6 to 8 are reserved for REM statements only and column 9 is allways a space for clarity. Executable code commences in column 10 only. Line continuation should be avoided wherever possible.

4. Header

Each program or procedure will commence with a standard header copied from a library containing a blank.

5. Program/Procedure names

Program names are restricted to 6 letters as follows:-

F.XXXX

F. denotes the final version of the program. Procedures are restricted to 9 characters as follows:-

_T360M012

where T360 is the program name
and M012 is procedure number 12

All procedures will therefore be of the form

```
DEFPROC_XXXM0YY
      .
      .
ENDPROC
```

6. Arguments

These are not generally necessary since most variables used are global ones.

7. GOTO Statements

These should not be used under any circumstances. Nesting should be used wherever necessary.

8. Loops

Nested loops should be indented by five spaces for each internal loop:-

```
FOR X = 1 TO 10
.....FOR Y = 1 TO 10
     .....A = X * Y
     NEXT Y
NEXT X
```

9. Arithmetic Statements

All expressions of the form A/B/C should have parenthesis inserted to clarify the order of evaluation.
All divisors should be checked to be non-zero. Implicit conversion in mixed arithmetic expressions should be avoided.

```
>LIST
 10REM **********************************************************************
 20REM *                                                                    *
 30REM *           BASIC PROGRAM STANDARD HEADER                            *
 40REM *                                                                    *
 50REM *           Task name     : F.XXXX                                   *
 60REM *           Module name   : XXXXMYY                                  *
 70REM *           written by    : Author                                   *
 80REM *           date          : dd/mm/yy                                 *
 90REM *                                                                    *
100REM *           Description   : dddddddd                                 *
110REM *                                                                    *
120REM *           Files access  : dddd                                     *
130REM *           Parameters    : dddd                                     *
140REM *           Called by     :                                          *
150REM *           Calls         : XXXXMZZ, XXXXMVV                         *
160REM *                         :                                          *
170REM *                                                                    *
180REM *           Mods          : mmmm                                     *
190REM *                                                                    *
200REM **********************************************************************
```

APPENDIX 2

POWERFAB/POLYTECHNIC OF WALES

Teaching Company Programme

P360 PROGRAM USER GUIDE

M. A. Bromfield
November 1986

CONTENTS

1. INTRODUCTION

This program allows the user to calculate the magnitude and direction of all the forces on the linkages of the Powerfab 360 digging arm. It uses a data file containing the major dimensions of the linkages (GEOMXXX) to carry out the calculations and outputs the force data to another data file (FORCXXX).

The user may select which units he wishes to work in (metric/imperial) and may calculate forces for a single or range of geometric configurations defined by the open, closed lengths and extension increments of each of the three hydraulic cylinders.

In addition to entering this data the user is also required to specify the magnitude and direction of the bucket tooth force. When the program is required to create a data file for the calculation of maximum tooth forces or stresses (using programs R360, D360) the user must input a unit tooth force.

2. REQUIREMENTS

The minimum system requirements are as follows:

1. BBC 64k Microcomputer
2. Single 80 track DD disc drive
3. Medium resolution monitor
4. Epson FX series printer

BUCKET TOOTH FORCE DATA

<1> BUCKET TOOTH FORCE [1 kN]
<2> ANGLE OF TOOTH FORCE TO TOOTH [20 deg]

For the calculation of maximum digging forces and stresses etc. a unit tooth force must be entered here; otherwise enter the actual desired tooth force in the correct units.
The same change procedure is used here and the program will only proceed once all the data enetered is satisfactory.

The next section of the program requires the user to enter the data files to be used for input and output of information. The input file GEOMXXX contains the dimensions of all of the rigid links for the structure. The output file FORCXXX contains the force data for use by programs R360, S360 and D360. The following menu appears:

PROGRAM DATA FILES

ENTER GEOMETRY DATA FILE NAME (GEOMXXX) ?_______

ENTER FORCES DATA FILE NAME (FORCXXX) ?_______

Ensure that the file names entered are correct otherwise the program may crash. After entering the file names the program will proceed and calculate the forces during which the following screen prompt will appear:

* * * PROGRAM RUNNING * * *

@@@@@ PLEASE WAIT @@@@@

Once all the relevant calculations have been made the program terminates and the following prompt appears:-

+++++++++++ PROGRAM TERMINATED +++++++++++

PROGRAM SPECIFICATION

Project	:	POWERFAB 360
Title	:	P360 PROGRAM SPECIFICATION
Author(s)	:	M.A.Bromfield
Version	:	0.01-
Created	:	3/12/86
Revised	:	
Ammendments	:	None
PROGRAM	:	P360

---

1. Name: P360

2. Function

This program allows the calculation of forces on all major components of the Powerfab 360 digging arm. The program uses an already defined data file GEOMXXX containing the dimensions of the major links on the digging arm. It outputs the calculated forces etc. to a user defined data file FORCXXX. The user may calculate the forces for a range of hydraulic ram lengths or a single configuration by setting the appropriate values in the geometric configuration data menu. The program may also be used to calculate maximum digging forces, the bucket tooth foce data menu valuse must be changed as neccessary. For both menus the default valuse are first displayed and then may be changed if required.

3. Environment

BBC 64K Microcomputer
Double-sided DD 80 Track disc drive
Epson FX85 printer
Microvitec CUB medium resolution monitor

3. USING THE PROGRAM

To start running the program first insert the P360 program disc in the disc drive (drive 0), hold down the <shift> key whilst pressing and releasing the <break> key. The following screen display will appear:

```
* * *  PROGRAM P360  * * *
--------------------------

PLEASE SELECT UNITS (M/I) ?_
```

If data is to be input in imperial then enter 'I' otherwise all calculations will be carried out in the default mode (metric). After selection of the units the following menu will appear:

```
GEOMETRIC CONFIGURATION DATA
----------------------------

<1>  BOOM CYLINDER CLOSED LENGTH             [545 MM]
<2>  BOOM CYLINDER OPEN LENGTH               [845 MM]
<3>  BOOM CYLINDER EXTENSION INCREMENT        [50 MM]

<4>  DIPPER CYLINDER CLOSED LENGTH           [545 MM]
<5>  DIPPER CYLINDER OPEN LENGTH             [845 MM]
<6>  DIPPER CYLINDER EXTENSION INCREMENT      [50 MM]

<7>  BUCKET CYLINDER CLOSED LENGTH           [545 MM]
<8>  BUCKET CYLINDER OPEN LENGTH             [845 MM]
<9>  BUCKET CYLINDER EXTENSION INCREMENT      [50 MM]

DO YOU WISH TO CHANGE ANY DATA (Y/N) ?-
```

If any data requires modification then reply 'Y', the following prompt appears:

```
PLEASE ENTER DATA ITEM YOU WISH TO CHANGE (1-9) ?_
```

Select the required options (1-9) as required and then press <return>; the next prompt will be:-

```
PREVIOUS VALUE WAS            X
PLEASE ENTER MODIFIED VALUE ?_
```

Enter the new value and the change procedure will be repeated until all data is satisfactory. The next menu to appear alows th user to enter the bucket tooth force data.

4. External References

Routines called

P360M01
P360M02
P360M03
P360M04
P360M05
P360M041
P360M042
P360M043
P360M044
P360M045
P360M046

Files accessed

GEOMXXX - linkage geometry data file (Input)

FORCXXX - forces data file (Output)

5. Structure

```
                 P360M00
                    !
                    !
                    !
 --------------------------------------------
 !             !             !             !
 !             !             !             !
 !             !             !             !
P360M01      P360M02      P360M03      P360M04
 !             !                           !
 !             !                           !
 !             !                           !
 ---------------                           !
       !                                   !
    P360M05                                !
                                           !
                                           !
                                           !
    ----------------------------------------------
    !        !        !        !        !        !
 P360M041 P360M042 P360M043 P360M044 P360M045 P360M046
```

Hierarchial Structure of the Program

6. Module Interface

P360M00 - Control routine for the program.

P360M01 - Set up geometric configuration data .

P360M02 - Set up bucket tooth force data.

P360M03 - Set up program data files

P360M04 - Calculate linkage forces.

P360M05 - Change data item.

P360M041 - Calculate geometric angles.

P360M042 - Calculate reactions.

P360M043 - Calculate bucket tooth forces.

P360M044 - Calculate bucket linkage forces.

P360M045 - Calculate boom forces.

P360M046 - Calculate balance of forces and moments.

7. Supporting Documentation

Technical notes may be obtained from the Teaching Company Progress Reports.

8. Test Plan

TBA

PROCEDURE SPECIFICATION

Project	:	POWERFAB 360
Title	:	P360M00 PROCEDURE SPECIFICATION
Author(s)	:	M.A.Bromfield
Version	:	0.01-
Created	:	5/12/86
Revised	:	
Ammendments	:	None
PROGRAM	:	P360
PROCEDURE	:	P360M00

---

1. Name: P360M00

2. Function

This program is the control module for the program P360. It coordinates the procedures for the setting up of data and the calculation of forces on the Powerfab 360 digging arm for a single or range of geometric configurations.

3. External References

Routines called

P360M01
P360M02
P360M03
P360M04

Files accessed

GEOMXXX - linkage geometry data file (Input)

FORCXXX - forces data file (output)

4. Processing Logic

Initially the graphics mode and data format are setup. Next the program heading is output to the screen. The user then inputs the required units for calculation (metric/imperial).
Procedure P360M01 is called to set up the geometric configuration data then procedure P360M02 is called to set up the bucket tooth force data.
Procedure P360M03 then sets up the program data files to be used and the relevant geometry data is read from file GEOMXXX. The user then selects whether or not force data is to be printed to screen for each geometric configuration. The output file is created and procedure P360M04 is called to calculate the forces and angles for every possible geometric configuraion. The file is then closed and the program termination prompt printed.

5. Program Design Language

```
START
     setup graphics mode and data format
     PRINT program heading
     INPUT required units
     PROC_P360M01 set up geometric config data
     PROC_P360M02 set up bucket tooth force data
     PROC_P360M03 set up program data files
     read in geometry data file
     INPUT do you wish to print data?
     PRINT program running prompt
     FOR all boom positions
          FOR all dipper positions
               FOR all bucket positions
                    PROC_P360M04 calculate linkage forces
                    output data to file
                    IF print requested
                    THEN print data to screen
               NEXT bucket position
          NEXT dipper position
     NEXT boom position
     CLOSE data files
     PRINT program termination prompt
END
```

6. Supporting Documentation

P360 Program Specification.

Technical notes may be obtained from the Teaching Company Progress Reports.

7. Test Plan

TBA

8. Reported Errors

There are no reported errors.

PROCEDURE SPECIFICATION

Project	:	POWERFAB 360
Title	:	P360M01 PROCEDURE SPECIFICATION
Author(s)	:	M.A.Bromfield
Version	:	0.01-
Created	:	5/12/86
Revised	:	
Ammendments	:	None
PROGRAM	:	P360
PROCEDURE	:	P360M01

---

1. Name: P360M01

2. Function

This procedure allows the user to view the default values used for the geometric configuration data for the Powerfab 360 digging arm.
The values may be modified as necessary.

3. External References

Routines called

P360M05

Files accessed

None

4. Processing Logic

Initially the default values of the gemetric configuration data are set up. Then the valuse are displayed on the screen. The user is prompted to change any item as neccessary using procedure P360M05 and the process repeated until all data is correct.

5. Program Design Language

```
ENTER
     setup gemetric configuration default data
     REPEAT
          display screen
          do you wish to change data?
          IF answer is 'Y'
          THEN PROC_P360M05 change data item
     UNTIL answer is 'N'
RETURN
```

6. Supporting Documentation

P360 Program Specification.

Technical notes may be obtained from the Teaching Company Progress Reports.

7. Test Plan

TBA

8. Reported Errors

There are no reported errors.

PROCEDURE SPECIFICATION

Project	:	POWERFAB 360
Title	:	P360M02 PROCEDURE SPECIFICATION
Author(s)	:	M.A.Bromfield
Version	:	0.01-
Created	:	6/12/86
Revised	:	
Ammendments	:	None
PROGRAM	:	P360
PROCEDURE	:	P360M02

---

1. Name: P360M02

2. Function

This procedure allows the user to view the default values used for the bucket tooth force data.
The values may be modified as necessary.

3. External References

Routines called

P360M05

Files accessed

None

4. Processing Logic

Initially the default values of thebucket tooth force data are set up. Then the values are displayed on the screen. The user is prompted to change any item as neccessary using procedure P360M05 and the process repeated until all data is correct.

5. Program Design Language

```
ENTER
     setup bucket tooth force default data
     REPEAT
          display screen
          do you wish to change data?
          IF answer is 'Y'
          THEN PROC_P360M05 change data item
     UNTIL answer is 'N'
RETURN
```

6. Supporting Documentation

P360 Program Specification.

Technical notes may be obtained from the Teaching Company Progress Reports.

7. Test Plan

TBA

8. Reported Errors

There are no reported errors.

PROCEDURE SPECIFICATION

Project	:	POWERFAB 360
Title	:	P360M03 PROCEDURE SPECIFICATION
Author(s)	:	M.A.Bromfield
Version	:	0.01-
Created	:	6/12/86
Revised	:	
Ammendments	:	None
PROGRAM	:	P360
PROCEDURE	:	P360M03

---

1. Name: P360M03

2. Function

This procedure allows the user to define the input data file (GEOMXXX) and the output data file (FORCXXX) for use by the program. The procedure reads in the geometry data file GEOMXXX.

3. External References

Routines called

None

Files accessed

GEOMXXX - linkage geometry data file (Input).

4. Processing Logic

Initially the program data file heading is displayed. Then the user is prompted for the geometry data file name (GEOMXXX), next the geometry data is read. The user is then prompted for the forces data file name for output of data (FORCXXX).

5. Program Design Language

```
ENTER
     display screen
     INPUT geometry data file name



     INPUT forces data file name
RETURN
```

6. Supporting Documentation

P360 Program Specification.

Technical notes may be obtained from the Teaching Company Progress Reports.

7. Test Plan

TBA

8. Reported Errors

There are no reported errors.

PROCEDURE SPECIFICATION

Project	:	POWERFAB 360
Title	:	P360M04 PROCEDURE SPECIFICATION
Author(s)	:	M.A.Bromfield
Version	:	0.01-
Created	:	6/12/86
Revised	:	
Ammendments	:	None
PROGRAM	:	P360
PROCEDURE	:	P360M04

---

1. Name: P360M04

2. Function

This procedure calculates the forces on the Powerfab 360 digging arm. Initially the geometry data is calculated then the reactions on the lower boom are calculated. Next the bucket forces are calculated and the bucket linkage forces. The boom forces and finally the balance of forces and moments on the dipper arm are calculated.

3. External References

Routines called

P360M041
P360M042
P360M043
P360M044
P360M045
P360M046

Files accessed

None

4. Processing Logic

Initially the procedure P360M041 is called to calculate geometry angles from the major linkage lengths. Procedure P360M042 is called to calculate the reactions at the base of the boom then procedure P360M043 is called to calculate the bucket forces. Next procedure P360M044 is called to calculate the bucket linkage forces and procedure P360M045 is called to calculate the upper boom forces. In order to check the results the balance of forces and moments on the dipper arm is calculated using procedure P360M046.

5. Program Design Language

```
START
     PROC_P360M041  calc geometry data
     PROC_P360M042  calc reactions
     PROC_P360M043  calc bucket forces
     PROC_P360M044  calc bucket linkage forces
     PROC_P360M045  calc boom forces
     PROC_P360M046  calc forces and moment balance
RETURN
```

6. Supporting Documentation

T360 Program Specification.
Technical notes may be obtained from the Teaching Company Progress Reports.

7. Test Plan

TBA

8. Reported Errors

There are no reported errors.

PROCEDURE SPECIFICATION

Project	:	POWERFAB 360
Title	:	P360M05 PROCEDURE SPECIFICATION
Author(s)	:	M.A.Bromfield
Version	:	0.01-
Created	:	6/12/86
Revised	:	
Ammendments	:	None
PROGRAM	:	P360
PROCEDURE	:	P360M05

---

1. Name: P360M05

2. Function

This procedure allows the user to view and modify items displayed in the current menu.

3. External References

Routines called

None

Files accessed

None

4. Processing Logic

Initially the user inputs the item number he wishes to change. The current value of the item is displayed and then the new value of the item entered.

5. Program Design Language

```
ENTER
     REPEAT
          INPUT item number you wish to change
          PRINT current values of item
          INPUT new value of item
          INPUT change another item? <ans$>
     UNTIL ans$ = 'N'
RETURN
```

6. Supporting Documentation

P360 Program Specification.

Technical notes may be obtained from the Teaching Company Progress Reports.

7. Test Plan

TBA

8. Reported Errors

There are no reported errors.

PROCEDURE SPECIFICATION

Project	:	POWERFAB 360
Title	:	P360M041 PROCEDURE SPECIFICATION
Author(s)	:	M.A.Bromfield
Version	:	0.01-
Created	:	6/12/86
Revised	:	
Ammendments	:	None
PROGRAM	:	P360
PROCEDURE	:	P360M041

---

1. Name: P360M041

2. Function

This procedure uses the data for link sizes to calculate the angles of the joints and the cartesian coordinates of the major hinge points on the Powerfab 360 digging arm.

3. External References

Routines called

None

Files accessed

None

4. Processing Logic

The geometric angles of the boom structure are calculated. Similarly the dipper and bucket geometric angles are calculated. Finally the cartesian coordinates of the major joint hinges are calculated.

5. Program Design Language

```
ENTER
     calcualate boom gemetric angles
     calculate dipper geometric angles
     calculate bucket geometric angles
     calculate cartesian coordinates for major joints
RETURN
```

6. Supporting Documentation

T360 Program Specification.
Technical notes may be obtained from the Teaching Company Progress Reports.

7. Test Plan

TBA

8. Reported Errors

There are no reported errors.

PROCEDURE SPECIFICATION

Project	:	POWERFAB 360
Title	:	P360M042 PROCEDURE SPECIFICATION
Author(s)	:	M.A.Bromfield
Version	:	0.01-
Created	:	6/12/86
Revised	:	
Ammendments	:	None
PROGRAM	:	P360
PROCEDURE	:	P360M042

----

1. Name: P360M042

2. Function

This procedure calculates the reactions on the lower boom section and mounting of the boom ram.

3. External References

Routines called

None

Files accessed

None

4. Processing Logic

The angle of the bucket tooth force to the vertical is calculated then the reactions are calculated.

5. Program Design Language

```
ENTER
    calculate angle of tooth force to vertical
    calculate reactions and angles
RETURN
```

6. Supporting Documentation

T360 Program Specification.
Technical notes may be obtained from the Teaching Company Progress Reports.

7. Test Plan

TBA

8. Reported Errors

There are no reported errors.

PROCEDURE SPECIFICATION

Project	:	POWERFAB 360
Title	:	P360M043 PROCEDURE SPECIFICATION
Author(s)	:	M.A.Bromfield
Version	:	0.01-
Created	:	8/12/86
Revised	:	
Ammendments	:	None
PROGRAM	:	P360
PROCEDURE	:	P360M043

---

1. Name: P360M043

2. Function

This procedure calculates the magnitude and direction of the forces on the bucket hinges for given applied bucket tooth force.

3. External References

Routines called

None

Files accessed

None

4. Processing Logic

The magnitude and direction of the forces are calculated.

5. Design Language

```
ENTER
     calculate angles and forces
RETURN
```

6. Supporting Documentation

P360 Program Specification.
Technical notes may be obtained from the Teaching Company Progress Reports.

7. Test Plan

TBA

8. Reported Errors

There are no reported errors.

PROCEDURE SPECIFICATION

Project	:	POWERFAB 360
Title	:	P360M044 PROCEDURE SPECIFICATION
Author(s)	:	M.A.Bromfield
Version	:	0.01-
Created	:	8/12/86
Revised	:	
Ammendments	:	None
PROGRAM	:	P360
PROCEDURE	:	P360M044

---

1. Name: P360M044

2. Function

This procedure calculates the bucket linkage forces and angles.

3. External References

Routines called

None

Files accessed

None

4. Processing Logic

The forces and angle of forces are calculated.

5. Program Design Language

```
ENTER
     calculate bucket linkage angles and forces
RETURN
```

6. Supporting Documentation

T360 Program Specification.
Technical notes may be obtained from the Teaching Company Progress Reports.

7. Test Plan

TBA

8. Reported Errors

There are no reported errors.

PROCEDURE SPECIFICATION

Project	:	POWERFAB 360
Title	:	P360M045 PROCEDURE SPECIFICATION
Author(s)	:	M.A.Bromfield
Version	:	0.01-
Created	:	6/12/86
Revised	:	
Ammendments	:	None
PROGRAM	:	P360
PROCEDURE	:	P360M045

---

1. Name: P360M045

2. Function

This procedure calculates the lower and upper boom forces and angles.

3. External References

Routines called

None

Files accessed

None

4. Processing Logic

The forces and angle of forces are calculated.

5. Program Design Language

```
ENTER
     calculate lower and upper boom angles and forces
RETURN
```

6. Supporting Documentation

T360 Program Specification.
Technical notes may be obtained from the Teaching Company Progress Reports.

7. Test Plan

TBA

8. Reported Errors

There are no reported errors.

PROCEDURE SPECIFICATION

Project	:	POWERFAB 360
Title	:	P360M046 PROCEDURE SPECIFICATION
Author(s)	:	M.A.Bromfield
Version	:	0.01-
Created	:	8/12/86
Revised	:	
Ammendments	:	None
PROGRAM	:	P360
PROCEDURE	:	P360M046

---

1. Name: P360M046

2. Function

This procedure calculates the balance of forces and moments on the dipper arm. This serves as a useful check for calculations on the entire digging arm.

3. External References

Routines called

None

Files accessed

None

4. Processing Logic

Initially the horizontal forces are calculated then the vertical forces on the dipper arm are calculated. Next the resultant moment on the dipper arm is calculated. The percentage errors are then calculated.

5. Program Design Language

```
ENTER
     calculate horizontal forces
     calculate error of balance
     calculate vertical forces
     calculate error of balance
     calculate moments
     calculate error of balance
RETURN
```

6. Supporting Documentation

P360 Program Specification.
Technical notes may be obtained from the Teaching Company Progress Reports.

7. Test Plan

TBA

8. Reported Errors

There are no reported errors.